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Contents for June, 1932

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McGRAW-HILL PUBLISHING COMPANY, INC., 330 WEST 42d STREET, NEW YORK, N. Y.

CHICAGO, 520 North Michigan Ave. 883 Mission St., SAN FRANCISCO Aldwych House, Aldwych, LONDON, W.C. 2

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Published monthly. $2 per year, 25 cents per copy. Central and South America, $3.00. Canada, $3.50, including duty. Foreign, $4.00 U. S. currency, or 20 shillings. Entered as second-class matter, January 17, 1930, at the Post Office at New York, N. Y., under the Act of March 3, 1879.

CABLE ADDRESS: "MCGRAWHILL, N.Y."

Member A.B.C.

NUMBER OF COPIES PRINTED THIS ISSUE, 9,500

WASHINGTON, National Press Building
PHILADELPHIA, 1600 Arch Street
CLEVELAND, 501 Guardian Building
DETROIT, 2-257 General Motors Building
ST. LOUIS, 1556 Bell Telephone Building
BOSTON, 1427 Statler Building
GREENVILLE, S. C., 1301 Woodside Building
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Encouraged by the editors of *Product Engineering,* C. A. Norman completes his tests and presents his conclusions on V-Belt life, overload and pulley diameter . . . *Page 248*

and in forthcoming issues . . .

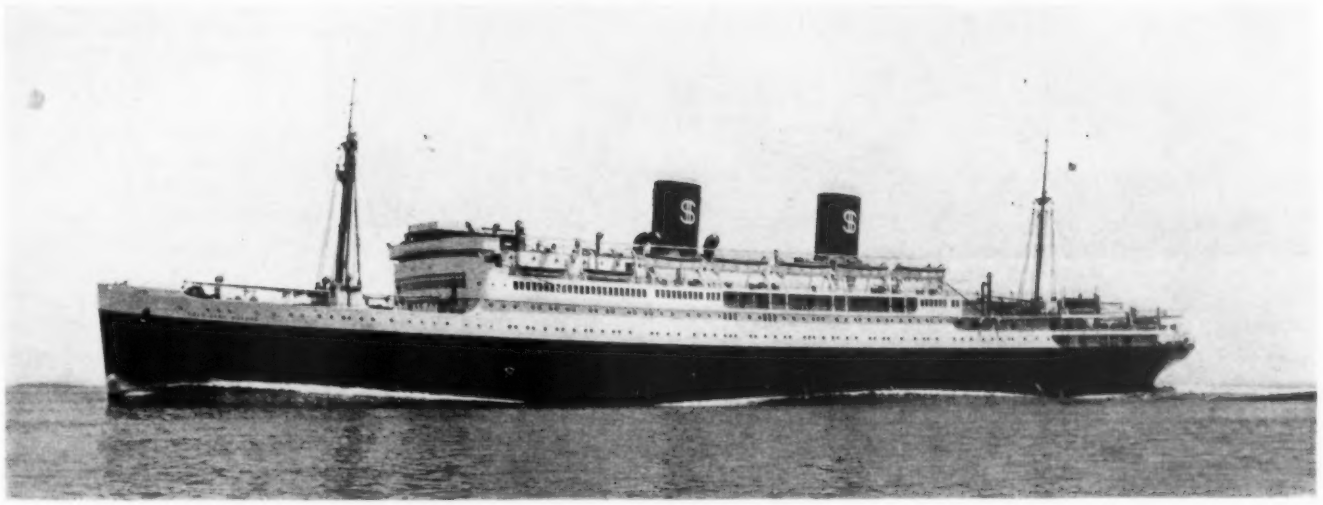
Dr. Irving J. Saxl will write on practical color selection. J. K. Olsen will continue his series on designing for economical steel stamping.

As scheduled for this issue but postponed through no fault of the author, A. C. Rasmussen will discuss the relation between heat-dissipating area and friction area on clutches and brakes

PRODUCT
ENGINEERING

MCGRAW-HILL PUBLISHING CO., INC. VOLUME THREE, NUMBER SIX

JUNE, 1932



The *President Hoover,* largest turbine-electric drive ship afloat, on her maiden voyage

Materials in Ships

W. E. BLEWETT, JR.
Assistant to the President
Newport News Shipbuilding & Dry Dock Company

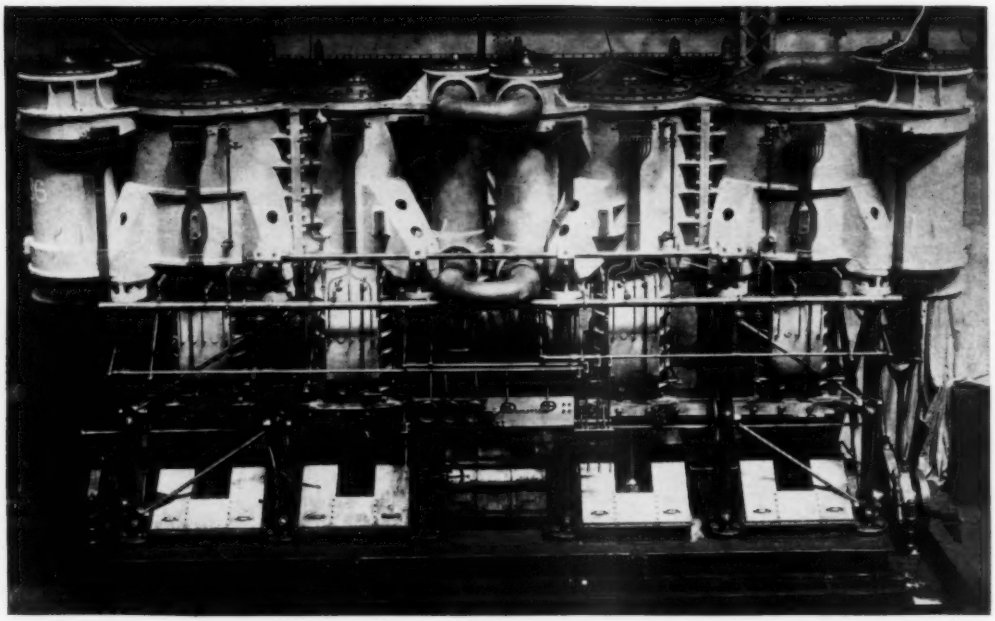
New, but untried, materials possessing characteristics desirable in ship construction are constantly being developed, but the degree of uncertainty connected with all of them makes the shipbuilder prone to hold to the materials whose reliability has been proved by practice. Perhaps he is not as progressive in this way as he might be, but ships are built to withstand severe tests of strength and durability, and the engineer feels that no unnecessary risks should be taken.

Mild steel has been most generally used for structural work and has proved successful. But we have now reached a period where we desire to construct larger ships which will have more speed and more carrying capacity per ton of steel. The same is true of ship machinery. We are using higher pressures and superheated steam with correspondingly higher temperatures, and the turbines, boilers, and auxiliaries have to be made to take care of this new condition. The modern ship also has an extensive refrigerated capacity, which brings before the shipbuilder materials which years ago he did not have to consider. Furthermore, the quicker turn-around demanded by steamship companies requires that freight handling facilities be bettered with respect not only to operating speed but also capacities.

▼

As the requirements of service become more severe and the number of available materials greater, the situation becomes more complex. It means that the engineer has to be versed in the latest developments in materials, and, most important of all, he must differentiate between sales propaganda and actual performance under service conditions

Then we have the subject of interior decorations, which are becoming more and more elaborate, with every new ship trying to surpass all others. However, it is the part of the ship that the average passenger sees, enjoys, and understands. Just what the ultimate outcome will be is problematical, but in an effort to insure the comfort of the passengers, interiors are yearly becoming more ex-



Reciprocating steam engines are by no means being entirely displaced by the steam turbine and diesel engine. Here is shown one of the latest designs of reciprocating marine steam engine without cylinder jackets. Note the elaborate provision for positive lubrication

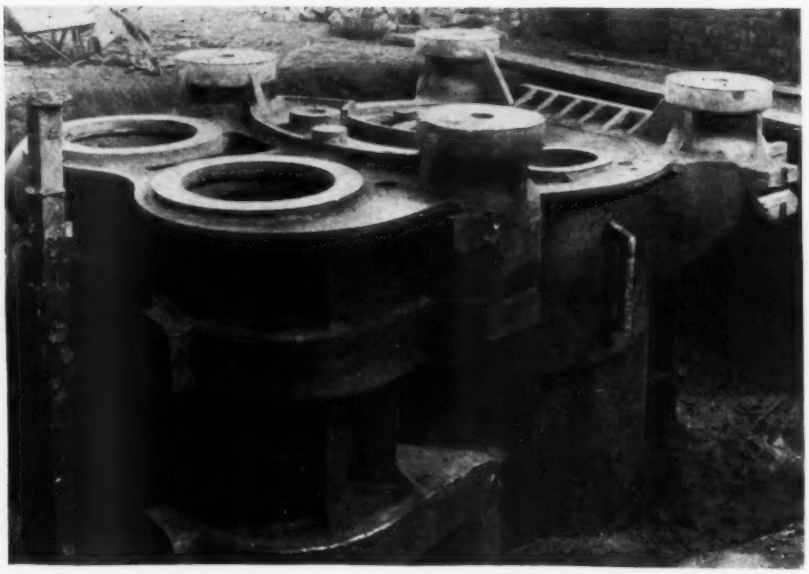
pensive. Not so long ago the interior decorations of a ship were made to conform to the structure, but now the tendency is to design the structure to conform to the interior decorations. With all the thought that is put into the finishing of public spaces and staterooms, one can well imagine the great quantity and variety of materials which have to be considered.

Exacting Specifications

The development in ships has taken place in the superstructures as well as in the hulls and machinery, and materials used are being purchased under more exacting specifications. In many cases, wooden hull parts are being replaced by metal; for example, window framings originally made of wood have in the past few years been made of brass and in the near future will probably be made of stainless steel or other similar metal.

With the rise in the cost of operating labor, more and more attention has been paid by the operators to upkeep, durability and utility, and ships will no longer appear with large quantities of polished brass to be kept clean.

In this country most ships are built under either Lloyds or American Bureau requirements. These requirements have proved satisfactory, but, though the characteristics desired of the materials are definitely stated, there is nothing to tell the manufacturer how he should obtain these physical characteristics. The Government also has its own specifications, which perhaps may be regarded as more rigid than the Bureau's. When the material is not covered by either of these, A.S.T.M. specifications are usually followed.



Engine castings are not only large but also involve complicated coring and many stiffening ribs

Materials used in ship construction probably have to be as carefully selected as those in any other construction work, and all factors have to be considered: weight, strength, corrosion, cost, workability, fire resistance, temperature coefficients and general appearance. Weight and strength are perhaps the most important, these being the basis of all the recent arguments for high tensile steels. Steel in a ship is put to one of the severest tests, that is, the continued reversal of stresses, and there is also the uncertainty of not knowing the exact loads to be carried by any part of the structure. In other words, figuring steels for strength in a ship is a different task from figuring stresses in a building. Weights have to be carefully investigated, particularly in superstructures. In addition to steel, of course, there are a larger number of other materials involved.

In ships the action of materials toward corrosion is a serious problem, as most parts are subjected to salt spray, or at best to sea air atmos-

phere. The hulls are at numerous times exposed to acid conditions in our rivers, as well as destructive bilge waters, oils and coals. Paints of various compositions are used for protection against corrosion, but paint is uncertain because the film is easily and frequently destroyed. There is also the problem of corrosion caused by stray electric currents or galvanic action between different materials.

The action of materials under temperature change has to be most carefully watched, particularly those materials going into the engine room. The hull of a ship is also subjected to severe temperature changes; for example, bunkering with fuel oil which has been heated.

There is no need to say much concerning fire-resisting properties, but if the marine organizations adopt a code of rulings similar to our city building codes, it will have a revolutionary effect on the design of passenger quarters and refrigerated spaces aboard our ships.

Structural Steels

The reaction of owners to high-priced materials is as to be expected. The operator wants to obtain as large a return from his investment as possible. He is not willing to spend extra money for materials when he can see no direct return. He thoroughly considers the appearance of the materials, for he has to have a ship that will be pleasing at all times to his passengers, but he also considers the amount of labor required for keeping his spaces presentable. The cost of upkeep may prevent the use of many materials. Finishes which may look the best may not stand the salt atmosphere, and more durable ones must be substituted.

The most important materials used aboard ship, and beyond a doubt the most discussed materials, are the high tensile steels. The use of these steels, however, cannot be considered a recent development, for high silicon steels were used in ship construction at least twenty years ago.

These steels may be divided into two general classes: (1) those obtaining high elastic properties through the use of alloying elements, such as carbon, vanadium, manganese, silicon and nickel; and (2) those in which the properties are obtained by heat treatment in one form or another, usually a tempering or a normalizing treatment obtained by controlling the temperatures during and at the conclusion of the rolling. Of these two classes, the alloyed steel is perhaps the most important and the one from which we may expect to obtain the best results, although as a rule they are more expensive because of the cost of the alloying elements. Each element has its effect on the structure of the steel, with results so complicated that it is a subject in itself for the metallurgist.

There are numerous high elastic steels. Those which we have recently experimented with for construction are tabulated with their physical properties in the following table:

Average Physical Properties of Some High Elastic Steels

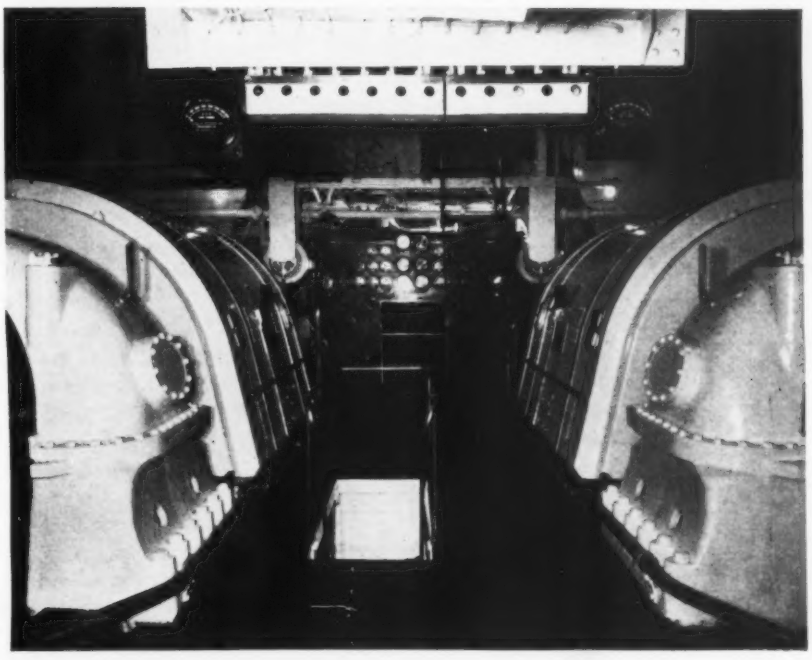
	Prop. Limit	Yield Point	Ult.	Per Cent Elon.	Per Cent Reduction of Area	Brinell Hardness
Navy high tensile	43,100	45,750	83,300	23.8	53	149
Low carbon manganese vanadium	59,500	72,300	99,950	18.7	59	159
Structural silicon	25,700	45,350	88,750	22	42	149
Structural silicon copper-bearing	43,700	48,150	79,650	25	52	143
Martinel	31,200	40,450	68,650	29	53	116
High tensile nickel ship	36,200	54,000	83,500	23	53.5	156
Mild structural	31,500	37,850	62,100	30	65.5	100

No tests were made to determine the fatigue limits of the various steels. In fact, we are rather dubious as to how much value can be attached to the fatigue figures, as the surface of the test specimen has such a great influence on the result. If the fatigue limits could be determined from what might be called practical, rather than laboratory surface finish, they would be of important value. Therefore, the best we can do is to assume a permissible bending fatigue limit of 40 per cent of the tensile strength and 50 per cent of that figure for the torsional fatigue limit.

There is also the compressive value of these steels to be considered. In a ship's structure a large majority of the members are subject to compressive loads, which means that, with the modulus of elasticity remaining the same, it will be necessary that the areas of the sections be maintained. Therefore little will be gained by the use of high elastic steel in purely compressive members.

An analysis of these high elastic steels indicates that any or all of them are adaptable to marine use.

Engine room of the *President Coolidge* showing the two Westinghouse 10,200-kw. main turbine-generator units with the instrument boards at the far end



All of them can be punched without heavy tool replacement with the exception perhaps of manganese-vanadium steel, but even this can be handled. However, the expense of working the high elastic steels is going to be greater than working our present mild steels until improved tools and methods of working, both hot and cold, are used.

Copper-bearing steel has often been considered for superstructure parts, such as ventilator cowls and deck houses, but we have not heard of it being applied extensively. From experiments I conducted with this steel there is apparently nothing to be gained by using it for underwater work. If used, it must be with care because, when in the presence of an electrolyte, galvanic action is likely to be set up in conjunction with other steels.

In addition to structural steel, many other steels are used in the construction of a ship. There are so many that space does not permit a discussion of all of them, but it will suffice to mention a few of the most important.

Non-ferrous metals take an exceedingly important part in the scheme of interior decorations

Forgings

Large hull forgings—namely, stern frames, stems, rudders, and such forgings, where fire welds are necessary—are made of 0.08 to 0.15 per cent carbon steel, with 0.3 per cent to 0.7 per cent manganese. This material has an elastic limit of 28,000 and 35,000 lb. per sq.in., with an elongation of 40 per cent. The forgings are used in their forged state and meet physical requirements without any heat treatment.

For small hull forgings, such as stanchions, hinges, cleats and rigging forgings, we use steel having 0.15 to 0.20 per cent carbon, and the same amount of manganese. This steel has an elastic limit of from 30,000 to 35,000 lb. per sq.in., and if normalized, its physical properties can be increased from 5 to 10 per cent. We also use this steel when case-hardening is required.

All heavy engine and hull forgings for merchant work, such as propeller line shafts, reduction gear shafts, coupling sleeves, crank pins and rudder stocks, are made from 0.20 to 0.30 per cent carbon steel having 0.4 to 0.7 per cent manganese. This steel, in practically all cases, is given a heat treatment which results in an elastic limit of from 32,000 to 45,000 lb. per sq.in. with from 25 to 38 per cent elongation.

For hydraulic work, large keys, diesel shafts, piston rods and crank pins, higher carbon steels are used. For large diesel engine piston rods, main rods, crank pins, and similar parts, carbon-vanadium forgings are used. Forgings made from this steel are used in the normalized and drawn state, giving an elastic limit of 58,000 to 65,000 lb. per sq.in. with an elongation of 24 to 30 per cent.

For propeller line shafts, rotor shafts and crankshafts, Government specifications call for nickel steel forgings containing $3\frac{1}{4}$ to $3\frac{3}{4}$ per cent nickel, with carbon content of 0.30 to 0.40 per cent which gives an elastic limit of 50,000 to 65,000 and elongation of 25 to 30 per cent. The addition of nickel to this steel has the result of improving its impact properties, increasing resistance to corrosion, and increasing elastic limit and hardness without loss in ductility.

Stainless steels, which are rapidly finding new applications in marine work, are now being used for turbine bladings, valve disks, and valve stems. Stainless steel has a useful place in galley equipment, as well as for fittings and ornamental work in passenger quarters. It is now used where non-ferrous metals were previously used, and we believe it is cheaper, less subject to corrosion, equally as stainless, and because it is stronger, it permits the use of lighter gauges. However, it is harder to work, which is a big disadvantage. But with succeeding jobs the shops and mechanics seem to understand it better, with a corresponding reduction in effort. As an austenite steel it will probably be used extensively in the structure adjacent to the navigating instruments, as it is supposedly non-magnetic. I have not heard of its use in this respect, but see no reason why it will not be successful.

The materials which are used for turbine bladings are numerous, depending upon the ideas of the individual engineer. On the first stages of the impulse blading we use stainless steel, on the intermediate stages monel metal, and on the last stages manganese bronze. For the diaphragm in the higher temperatures, we use stainless steel vanes cast into the steel, and in the lower temperatures monel metal vanes.

Rivet Materials

Materials for rivets going in the ship's bottom are always a source of argument, some marine operators preferring iron, others alloy, and still others plain mild steel. All these materials corrode and leak, but the rate of corrosion depends upon the protective coating and the care taken of the ship's bottom more than upon the material used. However, in determining the best material to use the service in which the ship operates should be considered. The proper heating and driving of shell rivets is also important, for good rivet material can be ruined easily by improper handling.

Monel metal and "stainless" steels predominate in the equipment of the galley, that is, the ship's kitchen

Non-ferrous metals and their alloys, because of their corrosion resistant properties, constitute a very important group of materials in ship construction, the quantities used being surprisingly large. For example, a ship of about 30,000 tons displacement contains approximately 350 tons of non-ferrous metals.

Because of its resistance to corrosion, copper is perhaps the most extensively used non-ferrous metal. There is an interesting condition relative to copper pipe. On the *California* and *Virginia* the copper pipe in the sanitary and salt water circulating systems showed severe pitting after a short time. This pitting is prevalent in the copper piping in many ships. On the *California* it was first thought this pitting was caused by stray electric currents, but the pipe lines in question were checked and no stray currents found. There was also a possibility of electrolytic action having been set up between the iron impurities in the pipe and the copper, with the salt water acting as an electrolyte, but this condition was never proved, as the copper used was seemingly pure. The pipe was replaced with red brass pipe, which is more difficult to work. But this pipe also pitted. On more recent ships, in these same systems, we are using genuine wrought iron galvanized pipe, and to date we have not received any serious complaints. I believe the Navy Department has also experienced considerable trouble along these same lines and is making rather extensive investigations of materials for these services.

Opinions relative to the use of condenser tube materials are very diversified. For most of our condensers, we have been using Admiralty metal, which contains 70 to 72 per cent copper, 1.2 per cent tin, 0.06 per cent iron maximum, 0.075 per cent lead maximum, and the remainder zinc. It is used principally because of its ductility and its resistance to corrosion. Although this metal has proved fairly satisfactory, in recent ships we have fitted our condensers with "Blackskin Finish" Admiralty metal tubes. The "Blackskin" is a finish of cupric oxide, which colors the tube black and serves as a protection against corrosion of the tubes. However, this material also has given trouble, and in a search for still better materials we are now using an aluminum bronze tube composed of 76 per cent copper, 22 per cent zinc, 2 per cent aluminum. It is still too early to judge its suitability.

The non-ferrous metals take an exceedingly important part in the interior decoration scheme. Window frames which were previously made of wood are now made of brass, and I do not think it will be long before they will be made of a stainless steel or other white non-tarnishable metal. The availability of extruded sections was a big factor in developing metal windows. Toilet fixtures and hardware, which used to be nickel plated, are now chromium plated, as the life of the chromium plate is longer. We are also rapidly finding out the finishes that will not stand salt atmosphere; for example, a silver finish will never hold up.

Although stainless steel has not been used extensively in interior decorations aboard ship, I see no reason why it should not replace some of the elaborate woodwork. There is a possibility of etching this steel so that it will accomplish a decorative as well as a structural purpose. But its property of transmitting heat and sound would be detrimental.

Steel Castings

There are a few locations in ships where high tensile steel castings are desirable; namely, rudder castings and stern frames. On Government work a casting with 85,000 lb. per sq.in. tensile strength, 53,000 yield point, and 22 per cent elongation is used, which properties are obtained by using either 1.5 per cent nickel and 0.6 per cent chrome or 3 to 3½ per cent nickel. The former casting is cheaper but more difficult to obtain, as shrinkage cracks cause a large number of rejections. These cracks are also prevalent, although to a lesser extent, in the 3 per cent nickel castings. For ordinary hull and machinery castings, an ultimate of 60,000 lb. per sq.in. and an elongation of 24 per cent is usually required.

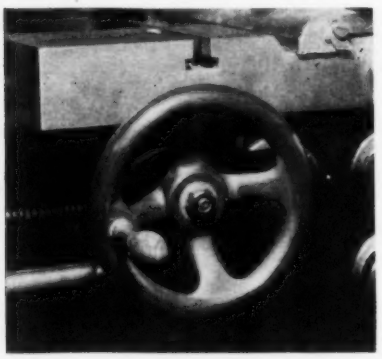
Chemical Composition of Some High Elastic Steels

	C	Mn	Si	P	S	Cu	Va	Ni
Navy high tensile	0.31	0.88	0.28	0.031	0.027	....	....	....
Low carbon manganese vanadium	0.14	1.42	0.20	0.019	0.017	....	0.18	....
Structural silicon	0.38	0.80	0.20	0.022	0.034	....	....	....
Structural silicon copper-bearing	0.33	0.70	0.30	0.026	0.028	0.24	....	....
Martinel	0.25	0.66	0.14	0.042	0.043	....	....	....
High tensile nickel ship..	0.21	0.82	0.24	0.042	0.022	....	....	2.71
Mild structural	0.19	0.85	0.11	0.027	0.031	....	....	....

PRODUCT DEVELOPMENTS

▸ New Material Uses

Nitralloy has been adopted for the cutter spindle of the new large Sykes gear generator. Bearing bushings are of alloy casehardened steel. This combination is expected to show no appreciable wear over a long period of years.



Molded from solid "Durez", this control wheel for a lathe weighs only 1.18 lb. as against 6.03 lb. for cast iron, and costs 15 per cent less. Other features are resistance to corrosion and pleasant surface

To reduce the effort required to change type magazines on its linotype machine, the Mergenthaler Linotype Co. is now making the magazine of aluminum alloy instead of brass. The new unit is 22 lb. lighter than the old one.

One-piece steel construction, spot-welded and Bonderized, then painted inside and out, is employed for the Kelvinator water coolers. Modernistic in design by reason of the vertical beads at the corners and the chromium-plated die-cast fittings, the water cooler's appearance is further enhanced because no screws or bolts are visible. Cooling units consist of silver-plated water reservoirs. Cooling is achieved by sleeve-type evaporators in which the refrigerant completely surrounds the water to be cooled.

Recent developments in the forging of Monel metal rods, shafts, and spindles in large diameters has made possible a further increase in the range of physical properties. A uniform yield point in excess of 70,000 lb. per sq.in. with an ultimate tensile strength of over 90,000 lb. per sq.in. on forged bars from 3 to 6 in. in diameter may now be obtained. Elongation value averages higher than 25 per cent. One application has been a 6-in. propellor shaft, 14 ft. long, weighing 1,534 lb. finished.

Hydrogen-peroxide, an electrolytic chemical used for bleaching purposes, is now to be shipped in an 8,000-gal. tank car built by General American Tank Car Corp. of formed and riveted aluminum plates.

Duralumin saves 1,500 lb. in the chassis and 2,000 lb. in the bullet-proof cab of trucks built by Ward-La France Truck Corp. for delivering silk. Side rails of the frame are built up from angles and plates, riveted. At the center the rails have a depth of 14 in. and at the axle 8 in.

Smooth glossy windings to which lint will not adhere are featured in the line of screenless open textile motors developed by General Electric. Large frame and shield openings allow lint to pass freely in and out. Dust- and oil-tight bearing housings eliminate lint-catching spots, which have always been found difficult to clean.

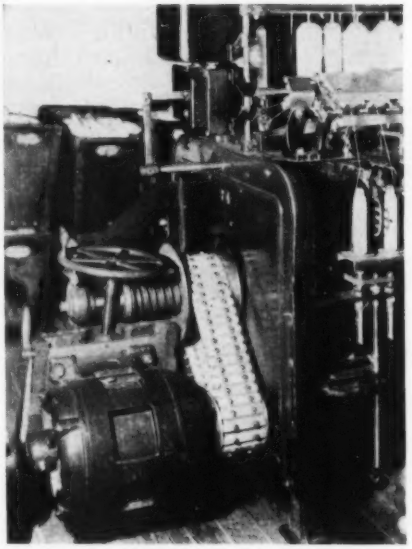
▸ New Applications of Parts and Units

For filtering out dust, dirt and cinders from the air entering its dining cars, the Louisville & Nashville R.R. has recently installed "Airgard" filters. These filters are compact cabinets, mounted one to a window in place of the shallow screen. Filter sheets are impregnated with special adhesive to trap dust particles. When air flow becomes restricted, the sheets are withdrawn and new ones slipped into place. Positive circulation of air is effected by two blower fans driven by one 1/30-hp. motor operating on 32 volts.

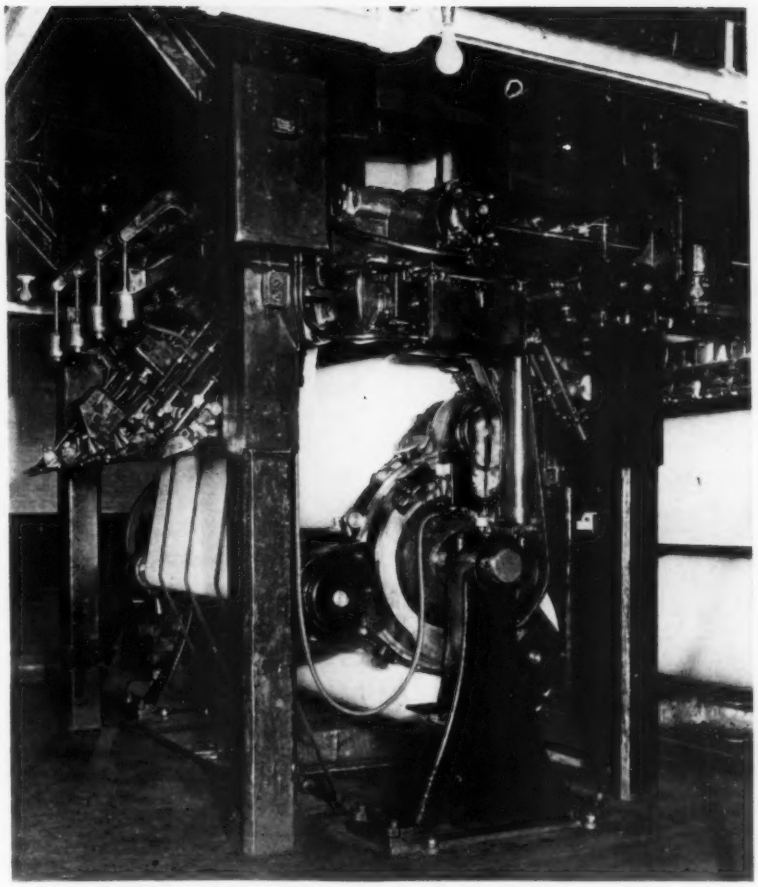
Hard-rubber-bushed bearings are fitted for register gears of the Model GA water meter developed by Worthington Pump & Machinery Corp. The train assembly comprises four cast bronze gears and four pinions, all fastened to Monel metal spindles. The disk or measuring chamber is in two sections, held together by a snap joint without screws. This construction permits the case to separate without distortion in the event of freezing. The stuffing box is cast on the main case instead of being a part of the train or register box, to eliminate leakage as no gaskets or locknuts are necessary.

Five-spoked camshaft gears of Phenolic Celeron invade the automotive field where solid web gears have heretofore held sway. Backed by extensive tests and 54,000 miles of service, Continental Diamond Fibre Company claims for them greater quietness, less heat, less weight and a safety factor of eight.

Roller bearings are used throughout the all-steel vibrating screen of the Telsmith portable crushing and cleaning plant. Practically no vibration is communicated to the screen supports. Special piston ring seals protect the bearings against dust and loss of lubricant.



Reeves Vari-Speed Motor Pulley, applied to a yarn twister frame, provides an immediate change in speed for different yarn sizes, an automatic increase in speed as the bobbin fills, uniform tension on the yarn throughout, more even yarn, less variation in size, and greater production. The drive, said to be equally applicable to warp and filling winds, is equipped with hand-speed adjustment and automatic-speed control



At the front end of the Wood double sextuple printing press installed at New York Times are seen the "Autoreel" and the "Autopaster." The latter device is at the upper left. It is an electrically controlled counterweighted mechanism that causes a brush to press the paper running off the expiring roll to the paste-daubed leading edge of the new roll. This construction eliminates slowing the press, which was necessary on earlier designs. The outstanding feature of the press in point of use is the fact that each unit is motorized, and a chain drive takes power to the main-shaft. Formerly, one main motor drove the entire group of units through a multiplicity of shafts and mitre gears

▸ Largest, Fastest Newspaper Printing Press

With capacity of 50,000 newspapers per hour, a new record, the Wood double sextuple press has just been installed by *The New York Times*. With full electric control, without stopping or slowing the automatic process of printing, folding, cutting and counting 14 complete newspapers per second, the "web" of paper is replenished. The press weighs 452 tons and utilizes 2,246 anti-friction bearings.

As one follows the newsprint paper through the cycle, the first new device he sees is the "Autoreel" mechanism. The device consists of two three-armed spiders mounted on a common shaft. Each pair of arms holds a roll of paper.

An integral part of the Autoreel is the "Autopaster," which is controlled electrically. Before a roll runs out, the pressman daubs the leading edge of the next roll with paste. When a control button is pushed, the new roll comes up to speed. A finger, following the thickness of the old roll, slips off at the proper point and closes an electrical contact. This causes a stiff brush to press the "web" from the expiring roll against the paste-covered edge of the new roll. The two webs go on together through the press at full speed. Immediately, a saw-toothed knife is caused by another electrical control to slash the web from the expiring roll. The "tail" and splice go through the press to the folder, where they and the six newspapers that contain them are ejected.

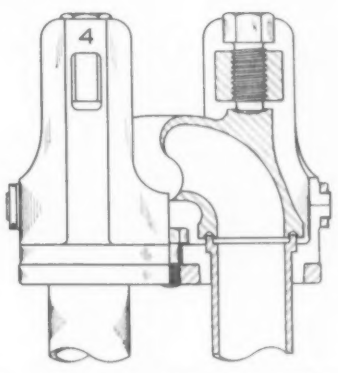
Drive to the various units of this press is furnished by the "Selsyn" system of self-synchronizing motors and Link-Belt chains. This type of operation prevents paper breakage. One motor is provided for each printing unit and for each folding unit.

▸ Ship Stabilizer

The possibilities of stabilizing large ships by means of a new type "Thyratron" control equipment, utilizing vacuum tubes and a small gyroscope, were demonstrated recently by Dr. E. F. W. Alexanderson, consulting engineer, General Electric Co. Using a small laboratory model, Dr. Alexanderson was able to imitate the rolling of a large vessel and show how his device would restore the ship to an even keel and keep it stable regardless of wave motion. His plan is the use of fins, designed like streamline rudders, attached to opposite sides of a ship, or using the force of gravity by changing of weights which may take the form of pumping water from one tank to another. In using the fins, these tilt up or down according to the roll of the ship, and their movement is actuated by a small gyroscope, which relays tiny impulses, greatly amplified by means of Thyratron control, through motors built inside the fins. Thus, if the ship should be inclined to roll to one side, the fins would immediately be tilted down on one side and up on the other as reversed conditions in the wave movement occur. The fins could be installed on existing ships without great expense. Theoretically, at least, they will not slow the speed of the ship to any appreciable extent.

▸ "Duplex" Sleeping Car

An experimental Pullman, called the "Duplex" car, with two single bedrooms on the "ground" floor and two more "upstairs," together with 10 standard sections is now in operation on the Pennsylvania. Each room



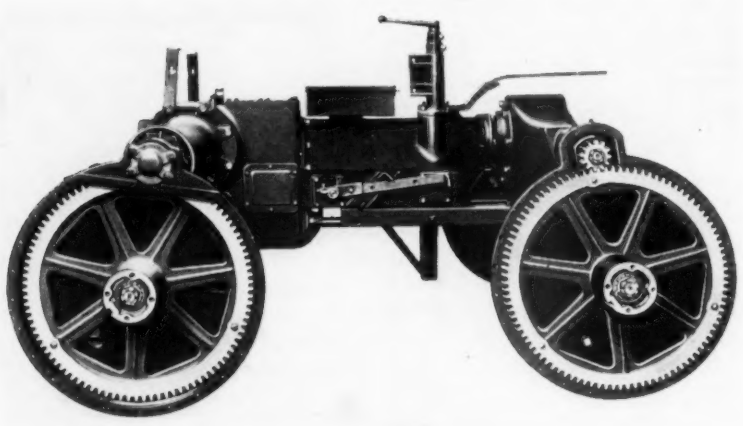
Quick replacement of tubes in petroleum cracking stills is facilitated by the design of this Stockham-Fantz return bend, Type SPR. This scissor-type of bend slips over the upset tubes, pulling them into place and acting as a pair of tongs

has a comfortable lounge for general travel, the back of the seat being turned down to form a spring-mattressed bed at night. In addition each room has toilet facilities, its own window, and self-regulating heating and lighting. With the Duplex room arrangement it was merely necessary to change the roof partially from a standard monitor to a turtle-back.

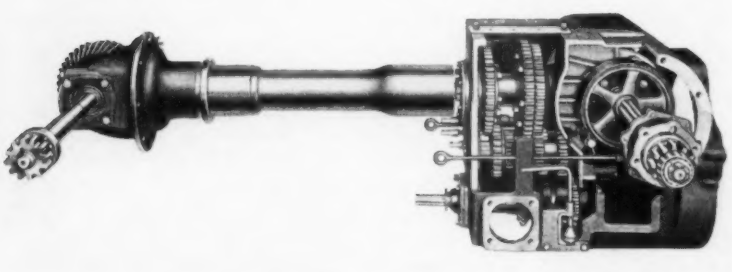
▸ Fire Hydrant Design

Whenever one sees a fire hydrant broken off by collision from a truck or automobile, he reflects that the repair will be laborious and expensive. Such fears are now relieved by the design of the Kennedy "Safetop" fire hydrant. The distinguishing feature of this design is that the safety breakable section is located a few inches above the ground line. Both the standpipe and the stem of the hydrant are in two parts which abut each other at the above section. The upper and lower portions of the standpipe are threaded at their abuting ends, and a correspondingly threaded "breaking ring" holds the two portions in rigid alignment. The upper and lower portions of the stem are similarly held firmly in proper position by a coupling which is secured to each portion by a large pin. Both the standpipe breaking ring and the stem coupling have a groove in line with each other at the level where the upper and lower standpipe and stem portions abut, and the entire assembly is so proportioned that the standpipe breaking ring and stem coupling form an inexpensive breakable section at a fixed point in the hydrant. Whenever a collision occurs, the entire top section of the hydrant carrying the nozzles detaches itself from the section in the ground without injury to expensive operating parts.

In the new Massey-Harris four-wheel drive tractor the front wheels are driven through universal joints, and the rear drive swivels. Consequently, all four wheels remain in constant contact with the ground



Hardened steel pinions mounted on adjustable Timken bearings drive the large ring gears at the wheels. The bearings are inclosed and run in oil. The motor is mounted on the pad at the left



Cut-away section of the driving mechanism, showing the drive and the location of the differential. There is a brake band at each side of the differential to aid in guiding the tractor. Thus, turning radius is 3 ft.

▸ Packard "12" Carburetor

Immediately below the two barrels of the down-draft carburetor used on the new Packard "12" are steel tubes, which, starting below the openings of the intake manifold, extend into the exhaust manifold. These tubes are conical at their lower ends. Any raw gasoline from the carburetor falls past the intake manifold openings and down into the conical portions of the tubes, which are heated at this point by exhaust gases. The gasoline is immediately evaporated and the vapor rises to the intake manifold. Jets located at the bottom of the tubes gather raw gasoline in cold weather starting and act as primers. Excess gasoline rises above the jets and is drained away to the ground.

▸ Fuel-Control Valves

In a fuel control valve for internal combustion engines, developed by Latham Co., Inc., the principle of operation depends upon synchronized vacuum power and electrical control so that necessary vacuum energy is not disturbed. The valve controls the inflow of fuel by setting up a zero vacuum at the carburetor butterfly during the time that fuel is not re-

quired, such as during deceleration. Control is obtained through a set of contacts which completes a circuit at closed throttle position. Current is obtained from the generator through a conventional circuit breaker and a solenoid. The solenoid allows the valve to close the atmospheric port of the diaphragm and also connects with the vacuum in the manifold. This is accomplished by a sliding valve arrangement which is spring controlled.

▸ Hydraulic Shock Absorber

Axial movement of a rotor-piston is the outstanding design feature of the McNab hydraulic shock absorber built by the Bridgeport Hardware Mfg. Corp. The rotor piston moves axially in the cylinder only when rotated by the means of the splined rotor shaft, which has a sliding fit through the center of the rotor. Axial movement is brought about by the square threads on the periphery of the rotor piston, and these engage corresponding threads in the cylinder wall.

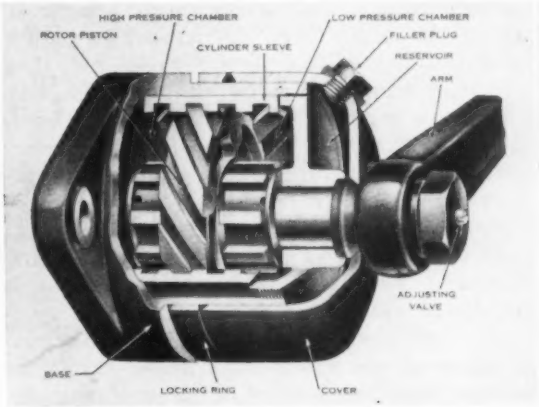
The hydraulic fluid passes through orifices in the rotor-piston body, as well as through the channels in the shaft. Resistance of fluid through these channels is governed by means of one-way ball check valves, and by the external needle valve adjustment. A spring-loaded relief valve is also incorporated in the piston to take care of excess pressure. Metering channels have a large area, as has also the rotor piston. Two-fold advantages result: first, high pressures are not built up, and second, the channels do not become obstructed. This design also prevents heating the fluid and changing its viscosity from effects of fluid friction, thereby obviating change in the operating characteristics of the shock absorber.

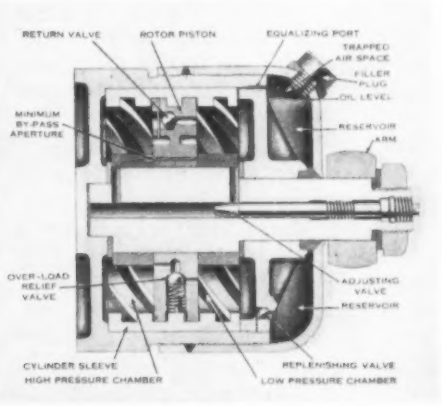
Frame height of only 21 in. from the ground permits mounting special bodies on this low-bed truck

Looking down at the truck frame behind the rear axle. Note outboard hanger for the spring suspension

▸ Low-Bed Truck

Specialized hauling operations, such as hauling plate glass, brick, tile, and garbage, which necessitate special body design, caused the Four Wheel Drive Auto Co. to develop a low-bed, front-wheel-drive truck. The frame construction of the Model LBU is unique in that it is made up of two sections on different levels and joined together by gusset plates. The drop in the frame is made at the rear of the cab. This type of frame construction, in conjunction with the rear axle mounting, gives a straight, clear loading space from the back of the cab to the rear of the frame. Height from the ground to the top of the frame is only 21 in., with the truck loaded. Full spring action is provided by means of four semi-elliptic springs. The front springs are mounted in the regular manner, whereas the rear springs are mounted on outboard hangers and attached to the drop-center rear axle by means of a special forging. This construction gives maximum ground clearance with full spring action.





In the McNab shock absorber, the rotor piston is moved axially by the square threads meshing with threads in the cylinder wall. The piston is free to move axially on the splined shaft, which is connected to the absorber arm. Channels in the piston body and in the shaft control the flow of the fluid, but are large so that high pressures are not built up

Motors and Their Controls

I. Direct-Current Apparatus

In this first of a series we consider the characteristics of the three types of d.c. motors, and the operation of manual controls. Succeeding articles will deal with automatic d.c. starters and specific applications, and finally a.c. apparatus

No thought can be given to the motor control until the motor has been selected, and this, of course, is determined by the kind of current that is available, whether a.c. or d.c., the nature and magnitude of the load, and speed requirements. Although there are relatively few d.c. motors being applied, in some industries they are used entirely to the exclusion of a.c. machines. For example, in copper refining plants where d.c. current is required for the manufacture of electrolytic copper, only d.c. motors are used. For the purpose of an analysis, it is also convenient to consider the d.c. motors first, as their electrical operation is relatively simple and easy to explain.

Selection of Motors

In the selection of a motor, the first factor to be determined is the required horsepower. Usually this will be governed by the horsepower requirements under running load, but in some instances the starting torque requirements might dictate the use of a motor larger than that required for the running load.

With reference to speed requirements, it must be remembered that a slow speed motor is larger and costs more than a high-speed machine. In some instances, price, performance or space might dictate the use of a speed reducer between the motor and driven machine in order to make possible the use of a high-speed motor.

In some installations, such as electrically driven contractors' equipment, a gas engine driven generator, located on the machine, furnishes the power. Under these conditions, the engineer is free to choose the operating voltage. Much can be gained by using higher voltages, this not only decreasing the size and cost of the motors but also the generator. Usually, 220 volts are used for such equipment.

Many machines require speed regulation. Either the speed must be maintained practically constant, or change of speed with load is permissible, or the motor and control equipment must be such as to make available regulated speeds within a definite range.

Direct current motors divide themselves into series, shunt and compound. Briefly, series motors are used when a very high starting torque is required and where constant speed is not necessary. The speed of the series motor varies with the load, and at no load the motor will "run away," causing the armature to burst due to excessive centrifugal stresses. For this reason, this type of motor should be used only when it can be geared or permanently connected to its load. Representative applications of series wound d.c. motors are for cranes, hoists and railway traction cars, and similar machinery where frequent operation is required.

The great drawback of the series motor is its lack of constant speed and the necessity of putting resistance in series with the armature in order to obtain speed control. The speed of the motor is approximately inversely proportional to the torque load on the motor over the normal operating range. For a given torque, the speed drop is directly proportional to the percentage drop in voltage occasioned by the insertion of resistance in the armature circuit. This resistance results in a power loss equal to the product of the square of the armature current times its external resistance. There is no way of circumventing this loss. For these reasons, it is evident that a series wound d.c. motor should not be used where constant speed is required or where the motor is to operate over long periods at a reduced speed. However, on applications where the operation under reduced speed is of short duration, the resistance losses may be far outweighed by the better operating characteristics of this motor for that particular service.

In all hoist applications, it is desirable that the light loads shall be hoisted as rapidly as possible, whereas the heavier loads should be hoisted at a lower speed. At the same time, the motor should be able to handle the heaviest loads for which the hoisting machine has been designed. The series wound motor is admirably suited to meet these conditions.

Shunt Field Motors

Shunt field motors are inherently constant speed machines, maintaining a practically constant speed under wide load variations. These motors are capable of developing a starting torque sufficiently high to meet the majority of applications. In addition to being constant speed machines, these motors also have close speed regulation, their speed being regulated by resistance inserted in the shunt field. It is merely necessary to bear in mind the fundamental principle that a shunt wound motor will tend to operate at such a speed that the sum of the volts lost in its armature because of the RI^2 drop plus the voltage of the counter e.m.f. induced in the armature will equal the impressed voltage. As the counter e.m.f. is solely a function of the speed and field flux, which latter is determined by the current density in the shunt field, it is evidently merely a question of controlling the strength of the shunt field in order to control the speed. Weakening the field increases the armature speed, whereas strengthening the field will cause the motor to slow down.

The compound-wound d.c. motor takes on the characteristics of both a shunt and a series motor and may be made to approximate more or less the characteristics of one or the other by making the ampere turns of the shunt or series winding more or less powerful proportionately. The two field windings can be connected so that the series winding either opposes or augments the shunt field magnetization. In the former, we have what is known as a differential compound-wound motor which, depending on the relative strength of the series and shunt field, may cause an increase in speed with an increase in load, resulting in unstable operation. For this reason, the differential compound motor is seldom used, and is not a commercial type.

In the cumulative compound motor, we have a strong starting torque, greater than that of the shunt motor, but the speed regulation is not as good. These cumulative compound motors are suitable for shears, punch presses, compressors, plunger pumps, rolls and similar applications where close speed regulation is not essential but a large torque is required at starting.

In fact, on such drives close speed regulation is not desired, since under widely fluctuating loads, the motor should drop off in speed when the peak load comes on, thereby transferring the load to the flywheel. The commercial general-purpose compound motor is of this type.

One fundamental difference between d.c. motors and a.c. motors that must be considered is the question of pull-out torque. This is the maximum torque that can be exerted by the motor armature. If the applied load torque is greater than this torque, the motor will stall. In the case of d.c. motors, the pull-out torque is practically unlimited if the line voltage holds up, while the short time overload capacity of the motor is generally determined by the load at which commutation becomes unsatisfactory. On the other hand, the a.c. squirrel-cage motor has a pull-out torque of probably 200 per cent of the full load running torque, while the starting torque is only 1.15 to 1.50 times the full-load running torque.

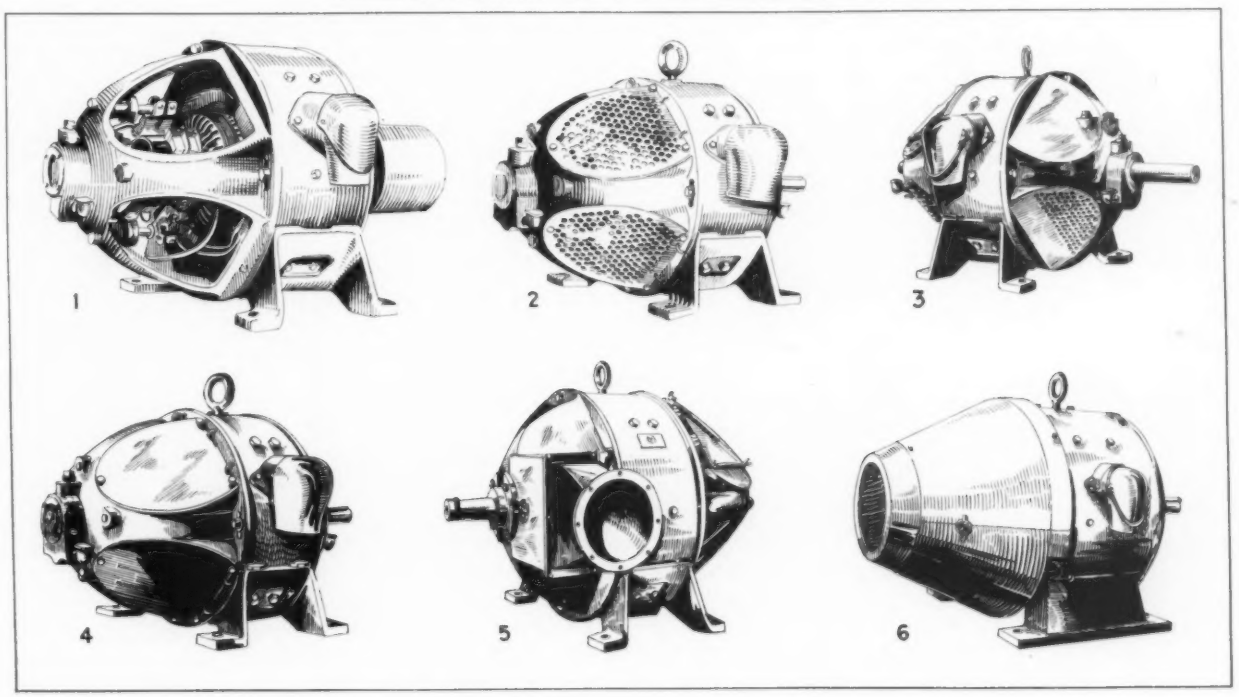
Effect of Inclosure

In the accompanying illustration is shown the approximate ratings of a given general purpose motor using different types of inclosures. The figures given are subject to wide variation, depending on the hp. and speed. On small motors or low speed motors, the percentage of normal rating that the motor can carry as a totally inclosed one will be greater because compared to the heat losses the radiating surface area is relatively large. Hence the heat dissipated by natural radiation from the frame is correspondingly relatively greater. However, not all open motors can operate continuously when totally inclosed, even under light loads. Hence, the question of inclosing motors should be taken up with the manufacturer.

Controls for series motors consist of an adjustable resistor connected in series with the armature and across the impressed voltage. The adjustable resistor prevents an excessive inrush of current at the time of starting. With the armature not rotating, there would be no counter e.m.f. induced, for which reason it is necessary to place resistors in the circuit. The same resistor may be used for speed control, providing it has sufficient current carrying capacity to prevent it from overheating. These resistors are usually separately mounted. For the larger motors, the resistance is in the form of cast-iron grids. Where reversing service is required, the control equipment accomplishes reversal by changing the direction of the current flow through the armature. A drum controller is usually used for cutting out the successive steps of resistance and for reversing the rotation.

For direct current shunt motors

Illustrating the effect of type of closure on the rated hp. of 10 hp., 230 v., 1,750 r.p.m. general purpose d.c. motor, (1) Open frame, 10 hp., (2) Semi-inclosed, 7½ hp., (3) Drip-proof, 7½ hp., (4) Totally-inclosed, 3 hp., (5) Pipe-ventilated, 10 hp., (6) Totally inclosed fan-cooled, 10 hp.



and compound motors, there are four fundamentally different types of manually operated rheostatic controls. First, there is the plain starting rheostat the purpose of which is to insert resistance in series with the armature when starting the motor to prevent an excessive inrush of current. The starting resistance is then cut out in steps. These starters have a low-voltage release which will open the circuit when the line voltage has dropped to approximately 40 per cent of the rated voltage. In the second type of starter, two rheostats are combined in one unit and are controlled from one handle. The one rheostat takes care of the resistance in series with the armature for starting, and the second rheostat regulates the speed by controlling the shunt field current. Usually, the maximum speed of the motor is limited to 400 per cent of that of the full field speed.

Limits of Speed Range

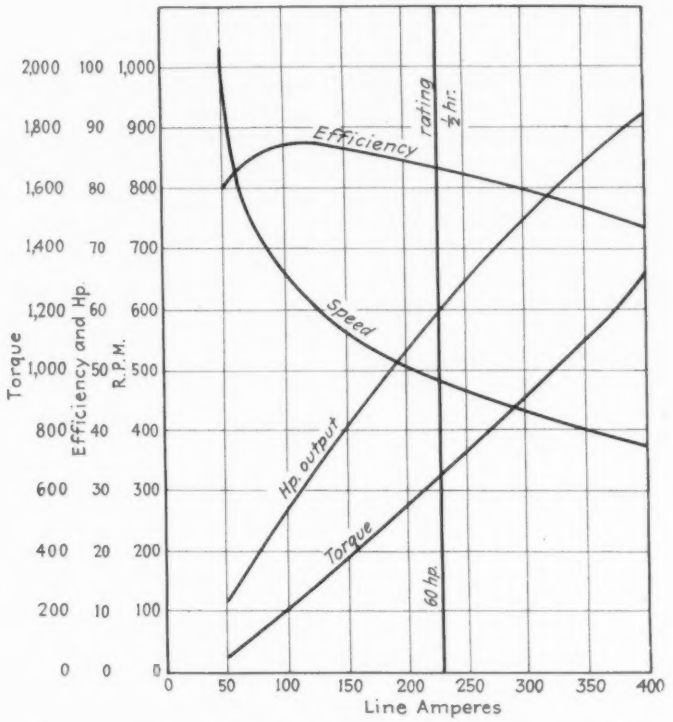
In the third type of d.c. motor starters, the control incorporates a rheostat in the armature circuit for starting and also for reducing the speed below the normal or rated speed of the motor. In this manner, a speed reduction down to 50 per cent of the rated speed can be procured. The fourth type of motor starter is the combination of shunt field and armature speed control. In this control, the maximum and minimum speeds are usually limited to 125 per cent maximum down to 50 per cent minimum, based on the rated speed of the motor. The 125 per cent maximum limitation is fixed by the design of the general-purpose d.c. motor, whose increase above the rated speed is limited by the design of the shunt field. The increased speed requires decreased field magnetism and hence a decreased field current. A condition is soon reached where the further weakening of the field will result in excessive sparking at the commutator and an unstable speed condition. On the other hand, the general-purpose d.c. motors that are specially designed for adjustable speed applications have their field so designed as to permit speed variations as high as 6 to 1, although 2 to 1, 3 to 1, and 4 to 1 represent the standard speed ranges. Thus, the full field or minimum speed of a $\frac{3}{4}$-hp. adjustable speed, general-purpose motor might be 600 r.p.m. and its maximum speed 1,800 r.p.m.

The combination of shunt and armature current control is seldom used on account of the RI^2 loss of power in the armature circuit when external resistance is inserted.

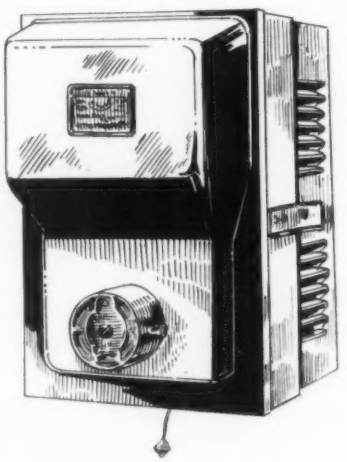
Drum reversing switches with separately mounted starting resistors or ballast resistors may be used for controlling small d.c. motors up to 10 hp. maximum for certain machine tool and similar applications when the motor is operated only intermittently and where low voltage and overload protection are not required. These drum reversing switches simply connect and disconnect the motor and reverse the flow of current through the armature for reversed rotation. This type of starter is used when no intermediate speed control is required and, incidentally, it can also be used for connecting small a.c. motors directly to the line without resistors.

These drum switches are manually operated, having but one point of acceleration, with the ballast or starting resistor in the circuit permanently. One of the types of reversing drum switches is a general service reversing drum switch having a cam contactor construction and complete individual magnetic blowout with arc splitters. This type of starter can be used for series, shunt, or compound motors up to about 10 hp. for standard reversing service. The resistor is separately mounted, and a protective panel may be added as extra equipment to fully provide overload and low-voltage protection for the motor when such conditions occur.

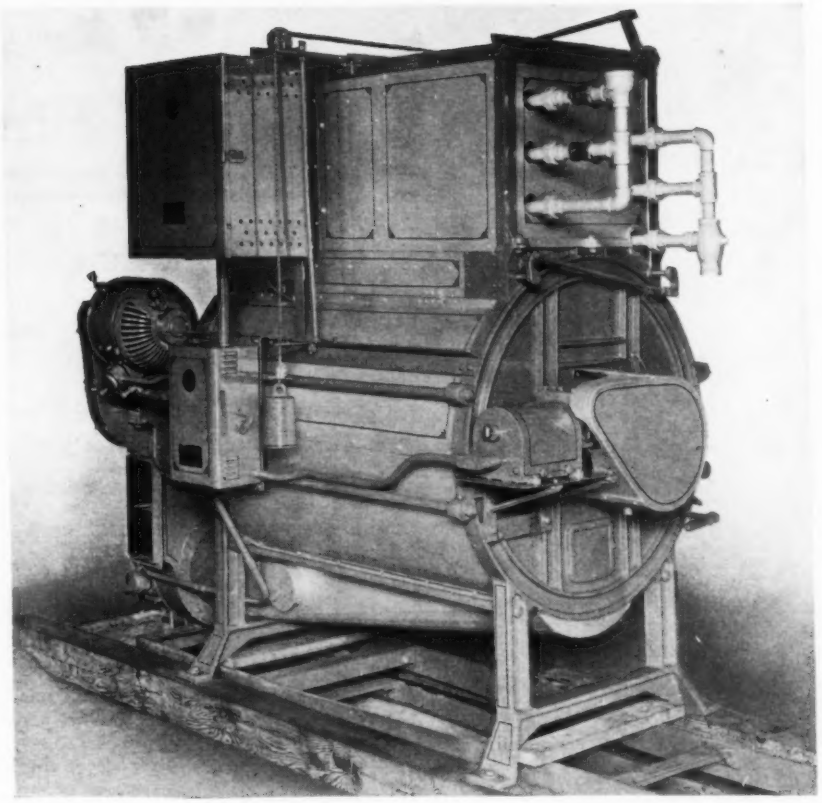
When a protective panel is used in conjunction with a general service reversing drum switch, it is necessary that the reversing drum switch be provided with a reset contact in



Characteristics of a d.c. series wound, 230 volt crane motor, showing clearly how the speed drops rapidly with increasing torque



Timestarter, wherein an adjustable clock escapement controls the rate of acceleration, with push button mounted on cabinet door



A laundry dryer shipped complete with two electric motors and controls built in. The larger case contains a d.c. reversing controller for the main motor, the smaller an automatic d.c. starter for the fan motor

the "off" position. With the drum switch in the operating position, there will be no external resistance in the armature circuit. Overload or voltage failure will cause the protective panel to operate, thereby disconnecting the motor from the line and stopping it. In order to make it impossible for the operator to throw the motor across the line with the drum switch in the running position and no external resistance in the armature circuit, the protective panel remains inoperative until the drum switch is returned to the "off" position, where a reset contact on the drum switch energizes the protective panel contactor, thereby closing it, the contactor being held closed by an electrical interlock. The drum controller can then be operated as desired.

Adjustable Speed Drum Controller

One type of drum controller for machine tools incorporates adjustable speed. Because of the unit mounting of the switch and the resistors, such controllers are quite compact in construction and can be mounted directly on the machine. The complete control is obtained with one operating handle which gives starting, stopping and speed control. This type of controller is used with adjustable speed shunt wound motors having speed ranges of $1\frac{1}{2}$ to 1; 2 to 1; 3 to 1; or 4 to 1. These controllers are reversing, but if desired the reversing feature can be eliminated by simply putting a mechanical stop on the drum to prevent the handle from being thrown into the position for reversed rotation. As these switches have a reset contact in the "off" position, overload and low voltage protection may be obtained by the addition of a protective panel.

Up to 5 hp., the starting and speed control resistors are mounted directly inside the controller. On account of the size of the resistor required for larger horse powers, it is necessary to have the resistors mounted separately.

In many machine drives, instantaneous stopping is required. One method is by the so-called dynamic braking wherein the armature circuit is short circuited through a resistor. Controllers that incorporate dynamic braking merely require the addition of the resistor and contact points for shunting the resistor across the armature circuit and disconnecting the armature from the line. Of course, the lower the resistance, the more rapid will be the braking. This feature of dynamic braking is used on planers and other machine tool applications.

Overload Protection

Protective panels are used in connection with manual controls for overload and no voltage protection. The electrical device for procuring the overload protection may be a thermally or magnetically operated relay. The thermal relay is merely a bi-metal strip, heated indirectly by the armature current. The increased temperature causes the bi-metal strip to bend and in so doing open a contact in the circuit of the main line contactor coil. This type of relay incorporates a time delay characteristic similar to the heating characteristic of the motor that it controls.

The magnetic type of overload relay depends for its action on its magnetic pull on a plunger. The pull is proportional to the armature current. Time delay is secured through the use of an oil dashpot.

It is evident that with the magnetically operated relay, the stronger the current is the quicker the solenoid will operate. The time required for tripping varies inversely with the amount of overload, a high overload current causing practically instantaneous action of the solenoid. Similarly, with the thermal relay, the higher the rate of the current, the more quickly the bi-metal element will be heated, causing it to bend and break the contact.

Low voltage protection is a fundamental characteristic of the line contactor by virtue of the fact that the holding coil releases the contactor which is pulled over by a spring when the voltage falls to approximately 40 per cent of the rated line voltage.

This by no means completes the diversity of control equipment for d.c. motors. Briefly, the types of starters and controls, additional to those described above, include general service reversing drum switches and magnetic starters for remote control, each of which is available in a variety of modified designs and in combination with various accessories such as magnetic brakes, preset speed regulation, limit swtches, and similar units. These will be dealt with in detail, together with specific applications, in the next article of this series.

Taper Pin

F. A. FIRNHABER
Machine Designer
Landis Machine Company

For relatively light loads, taper pins have the advantages of simplicity and ease of disassembly, both when used in place of keys and when used as dowel pins to locate members. Here are some typical applications and data on the holding power of taper pins as determined by tests

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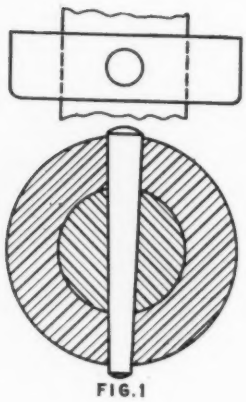


FIG. 1

Fig. 1—The conventional, familiar way of applying a taper pin, which depends upon friction to hold the pin in place. In gearboxes and other sealed mechanisms where it would be injurious for a pin to work loose, positive locking means must be provided

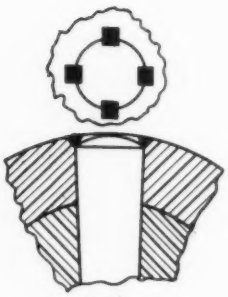


FIG. 2

Fig. 2—The large end of the pin comes just below the surface of the external member it is holding and is staked as shown in the plan view. These little swellings or burrs straighten out or shear off if it is necessary to remove the pin, but usually will score the surface of the pin. It should be noted that cast iron does not stake readily as it is brittle and will not flow

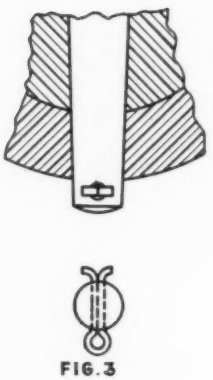


FIG. 3

Fig. 3—A small cotter pin retains but does not prevent loosening of the taper pin

Fig. 4—With this design, the taper pin is pulled tight with the hex nut which bears against a flat on the external member, although this flat is not necessary. Some engineers prefer to use a lockwasher under the nut, in which case both the nut and the external member should not be hardened, to permit the lockwasher to get a grip

Fig. 5—In this design the screw stud is expanded and locked by the use of a taper pin. The stud is slitted as shown in end view. The taper pin rests in the bottom of the hole and the stud is screwed in until it can be turned no farther

Fig. 6—Here the pin is flattened off and used as a wedge. This method calls for accurate work, but if the hole is reamed too large or wears too much the next larger taper pin may be used

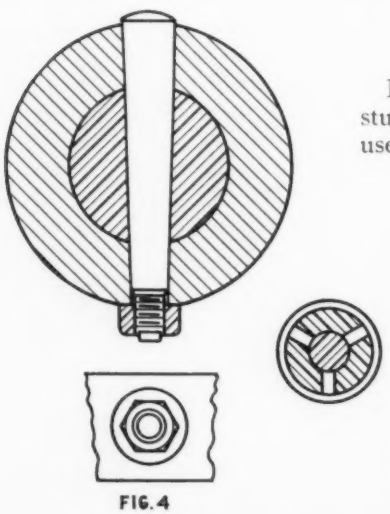


FIG. 4

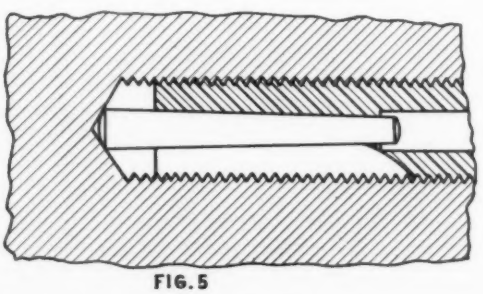


FIG. 5

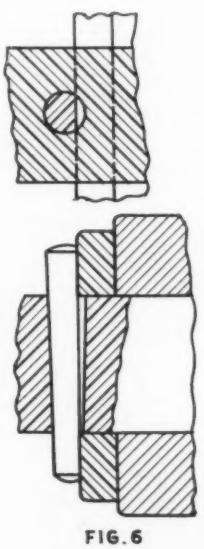


FIG. 6

Applications

Figs. 7, 8, 9—When a taper pin is to be used in a blind hole, one of the three methods shown here can be used. To facilitate loosening the pin, a square may be milled at the large end as in Fig. 7. It is well to cyanide this squared end. Fig. 8 shows a special form with a square head, the flat of which is equal to the large diameter. This type should be hardened all over and ground on the body. In Fig. 9 the pin is threaded and jacked out by a hex nut against a washer. The top end should be cyanided so it will not be pounded over during assembly. A fine thread should be used so as not to weaken the pin by too small a root diameter. For appearances, the washer and nut are left on, but this does not render it foolproof. This form is used as a dowel pin where the held member must be located accurately

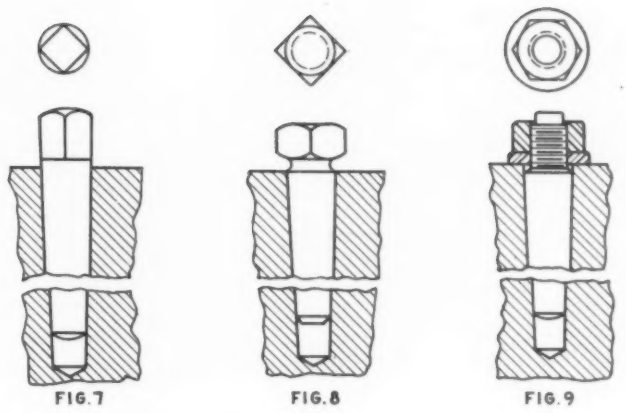


FIG. 7 FIG. 8 FIG. 9

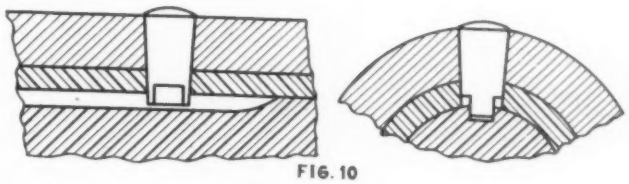


FIG. 10

Fig. 10—This shows a twofold purpose. The sawed off taper pin acts as a holding device and a key guide to the slidable inner member

Figs. 11, 12, and 13—Some time ago, the holding power of taper pins was tested to see which of the three methods shown here is the best. A No. 6 taper pin was driven in place as in Fig. 11, directly on the center line of the $1\frac{3}{16}$-in. shaft. In Fig. 12 two No. 4 pins were placed $1\frac{3}{16}$ in. apart, with half of the pin in the external member and half in the shaft at the horizontal center line. In Fig. 13, two No. 4 pins were placed $1\frac{5}{16}$ in. apart, where the full diameter of the pins passed through the shaft on the horizontal center line. In all three cases the shafts were the same diameter and of the same material, the external member being a cast-iron lever. The area of the No. 6 pin at the center line of the shaft is 0.077 sq.in. and that of a No. 4 pin is 0.039 sq.in., or practically half as much. Therefore, the area of one No. 6 pin and two No. 4 pins are equal

In testing for torsion or twist it was found that in Fig. 11, the No. 6 pin sheared under a load of 1,250 lb.

Fig. 12 was loaded to 1,150 lb., when the casting broke approximately at the outside diameter *A*

In Fig. 13 a load of 1,325 lb. was reached when the casting broke through the hub around the shaft

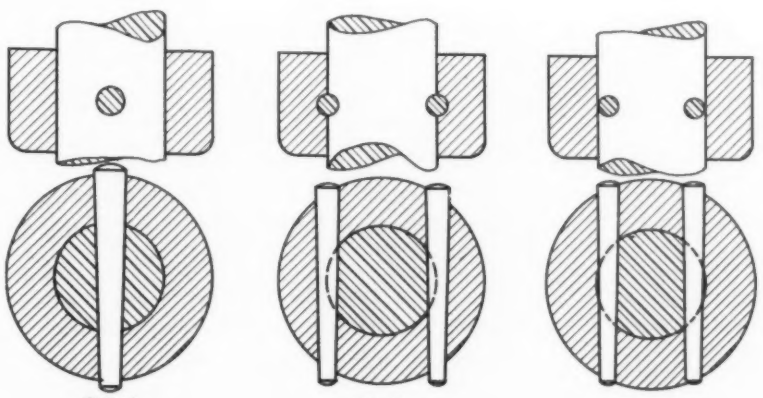


FIG. 11 FIG. 12 FIG. 13

It was further found in examining the smaller pins in Figs. 12 and 13, that only one pin carried the load. This showed conclusively that two smaller pins placed tangentially to the shaft, either at the center line or inside the shaft, Figs. 12 and 13, are over twice as strong as the larger single pin, Fig. 11, and that one smaller pin placed as in Fig. 12 and 13 is more than equal to the larger pin. It will be understood that the duty imposed upon the shaft, lever and pins during the natural functioning of the machine would never attain that reached in the testing machine

EDITORIALS

KENNETH H. CONDIT
Editor

"Where Is It Used?"

In the lead article of this number of *Product Engineering,* W. E. Blewett, assistant to the president of Newport News Shipbuilding & Dry Dock Company, frankly states, "The shipbuilder is prone to hold to the materials that he knows from past experience have proved reliable." Every designing engineer will murmur a solemn "Same here," to that statement, while manufacturers of materials and finished parts will look upon it as further evidence of the ultra-conservatism of the engineer. Indeed, in this respect the engineer is sometimes accused of blindly hindering progress, an accusation which will be found incorrect when the whole situation is analyzed.

The progressiveness of the engineer, as evidenced by his many improved products, new constructions, and the development of complicated mechanisms and machines, is beyond question. In these phases of his activities, the engineer shows a spirit of daring pioneering that has no equal in any other profession. Then why is it that this same spirit apparently is not exhibited by the engineer when he is faced with the question of specifying a new material or part?

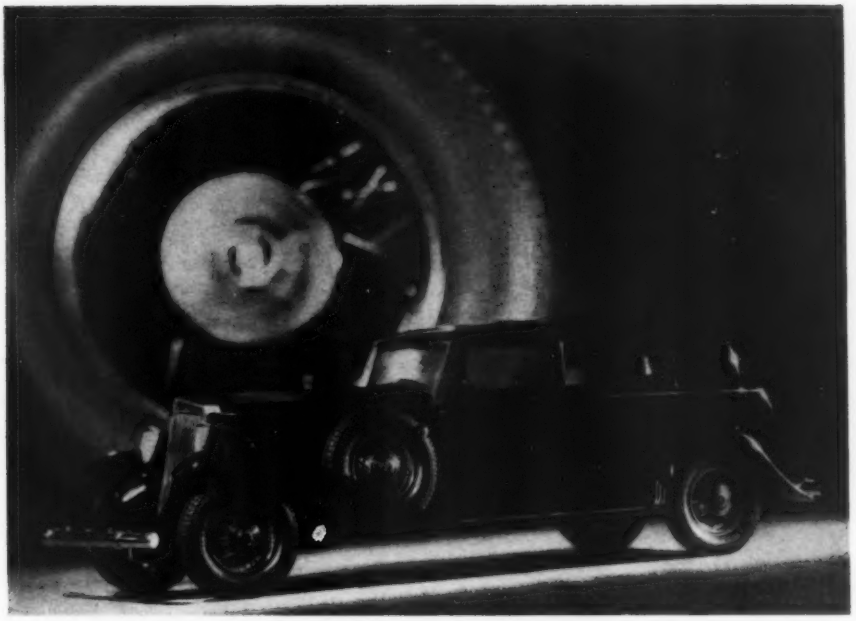
Manufacturers of parts and materials do not seem to appreciate the fact that in the development of new designs and mechanisms the engineer usually can calculate his loads and stresses. If some of the factors entering into the design are indeterminate, the engineer draws upon his past experience, or resorts to test. In an extreme case, he might be compelled to make a guess, but the guess would be an intelligent one, based on wide experience. He makes load, speed, stress and vibration calculations with tried and trusted formulas; he has faith in the result, even though he be venturing into an entirely new construction. If the calculations are wrong, he has no one to blame but himself.

But when it comes to approving the use of a new material or finished part, the engineer is treading on unknown ground. Unless he can ascertain the results of the use of the material in question on similar applications, he has nothing on which to base his judgment. He looks upon sales arguments as being over-enthusiastic; in many instances, he considers the results of laboratory tests open to suspicion because test conditions rarely simulate actual conditions in service. Engineering formulas and slide rule calculations cannot be applied, and because the engineer bases his decisions on hard facts, the sales and advertising claims that appeal to sentiment rather than reason fall on barren soil.

Manufacturers should take their cue from the first question that the engineer almost invariably asks when he is approached by the salesman offering a new part or material. "Where is it used?" To answer this searching question manufacturers should devote more of their advertising space to featuring new and successful applications of their products. They will find it profitable to give the widest possible publicity to new applications for their materials or parts.

Technical publications catering to the designing engineer have a basic duty to perform in this respect. Their editors must be vigilant in reporting new uses of materials and new applications of finished parts, in order that their readers may keep abreast of progress. *Product Engineering* has always catered to this particular need. As a special service to readers, there are listed every month in the "Product Development News" department of this paper, applications of new parts and materials that have proved successful. These items are gathered by editors in the field and by special contact with several hundred manufacturers of parts, materials and machines.

The engineer is not conservative; he is progressive. But he has a highly commendable habit of basing decisions on facts rather than generalized claims. It is up to the manufacturers and their sales representatives to spread the facts regarding their wares. And the most powerful sales argument is a fitting answer to the engineer's insistent question, "Where is it used?"



"First rough sketches, then careful drawings. Then a sixth-scale model was built, perfect in every respect"

The Artist in Industry

What He Does and How He Works

WALTER DORWIN TEAGUE

You cannot permanently suppress an instinct. Mechanical marvels have grown commonplace; our eyes have grown critical. We are reasserting our ancient right and instinct to believe that a thing cannot really be right unless it *looks* right. So the artist has been drafted by the manufacturer and the engineer to supply to the Machine Age that care for form and appearance which was the special pride of the old time handcraftsman. Not an easy job.

It is a new profession, demanding a special technique and special equipment, although time was when it was taken for granted that if a man was a good artist he was also a good engineer. Cellini, Michelangelo and a crowd of smaller fry were called upon as a matter of course to make fortifications or war-engines as well as palaces or pictures. Da Vinci invented the wheelbarrow as well as the Mona Lisa. But the time is probably past when any man is big enough to be so inclusive. If artist and engineer can each be good in his own field both will be busy.

But they must be able to do team work. The artist must rely on the engineer at every turn, work with him, think in his terms and share his viewpoint. That is, if he is equal to his job, he will be the kind of artist who thinks in practical processes and concrete results; he will understand machines and techniques, and have a wide knowledge of production methods. He will know that the best place to design a product is in the factory where it is to be produced, that tools cost money and that an unnecessary operation is just so much waste. He will accept these conditions cheerfully, and realize that his designs are not of the slightest value unless they can be *made,* and made economically.

The merely visionary plays no part in either the work or the conversation of this kind of an artist. If, as Mr. Lee says in his recent article, (*P.E.*—Vol. 3, page 201) the engineer sometimes finds himself unable even to understand the artist's language, the engineer is simply unfortunate in the artists he has met. Let me assure him they are not all like that.

Granted that the artist is able to work with engineers successfully, just what capital of his own does he bring to the partnership? First of all, and quite naturally, he is expected to contribute a sense of design. Briefly, this may be described as a habit of seeing things not as specific objects but as masses and silhouettes, lines and surfaces, patterns and arrangements; it implies an eye sensitive to line, form and color.

In addition the consultant should have a dash of that creative ability which carries him a little further than these rules of the game indicate. If

he is really an artist, he possesses the power of "visualization": that is, he can see quite clearly in his mind things which do not yet exist, and know in advance how a project will look when completed.

Broad Background Essential

The design consultant should be equipped, also, with a broad and diversified experience. Since he works in a wide range of industries, learning lessons in each to be applied in the others, he should bring to a specific problem a background of experience and a considerable variety of resources, plus a fresh, outside, and not too technical viewpoint. When a man has lived with a single product day in, day out, for years, it is difficult for him to see that product in any but its familiar guise. The consultant will be able to see it in an entirely new light, and often to recast it in more effective form. In doing this, his experience often enables him to effect improvements and economies. He may, for instance, plan his design so as to eliminate unnecessary parts and operations. Or he may cut a cost where it is doing no good, and apply the money saved where it will directly enhance eye-appeal.

Products must not only be *made*, they must be *sold*. The design consultant therefore is expected to have a comprehensive knowledge of style, style being nothing more nor less than the way people in general at a given time like things to look. If you make your product that way, these people are much more likely to buy it than if you disregard their preferences. The designer must be sensitive to trends, know how the spirit of his times is expressing itself, and be able to predict with reasonable accuracy the trend which taste will take in the immediate future.

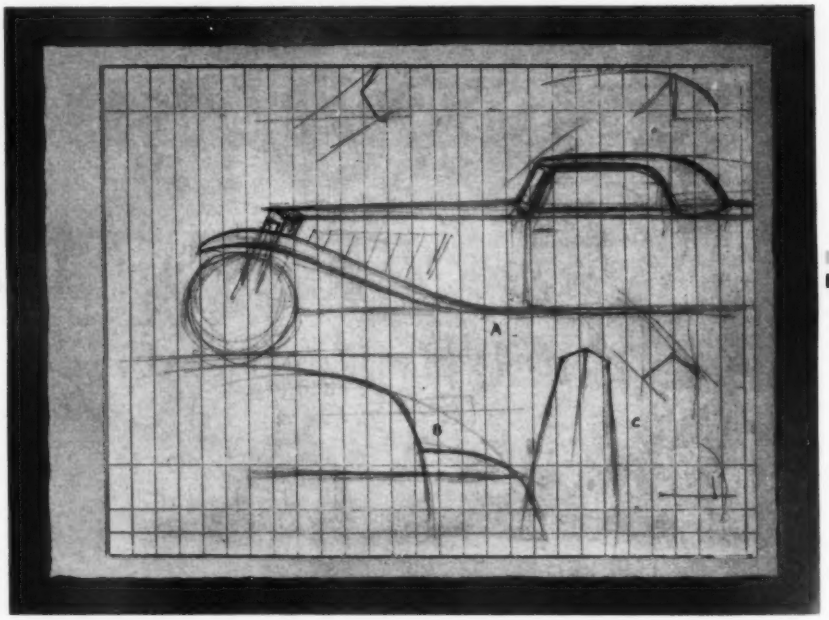
The product may be almost anything: a gas boiler or an electrical stoker, a weighing machine, a glass jar, an industrial thermometer or an automatic recorder, an automobile or a calculating machine, a drip coffee pot or a typewriter. The consultant brings to its fabrication his trained sense of design, his knowledge of materials and processes gained in a broad industrial experience, and his knowledge of popular taste. He must go into the plant where it is to be made, learn from the engineers its practical requirements, its limitations, and the facilities of the plant, study costs of tools, operations and materials. From the sales manager and the advertising manager he will learn its market, its competition and the method of introducing it. Out of all this information a picture of how it ought to look gradually crystallizes in his mind. Often this is surprisingly different from how it does look.

Cooperation Essential

When this stage is reached, and not before, drawings may be made, and models. Step by step the new design is worked out to eliminate any possible "bugs"—impractical features, excessive costs, difficult operations. Often a series of adjustments and revisions must be made, artist and engineer working the design over together until both are satisfied with the result. In this process the sympathetic cooperation of artist and engineer is absolutely essential to success.

A good design always grows from the inside out. Just as our bones and muscles determine the form of our bodies, so the mechanism and the function of any manufactured article should determine its outward appearance. I have said elsewhere that "a good designer never adds anything but beauty to the thing he designs." By making it look its part and tell its story candidly he can give it that appearance of high efficiency we admire in all thoroughbred organisms. Mere candor of course isn't enough; an anatomical chart is not our idea of a beautiful body. But starting with sound bones and muscles, the designer can model a form that will have genuine appeal. His work is mainly a careful adjustment of lines and proportions, surfaces and masses, a shrewd choice of materials, and an organizing of details into a unified whole. He has learned from his study of popular taste that he can get further today with a well-proportioned simplicity, cannily accented, than with any amount of elaboration.

An automobile offers one of the most difficult problems in design today; production, efficiency, cost and style are all vital factors. Thus it may be worth while to review the



Facsimile of first sketches for the Marmon Sixteen, made during the conference at which Mr. Teague was informed that the motor was to be a 200-hp. job. Almost all the basic features of the design were then laid down. Two months of study, however, preceded these sketches, followed by careful drawings, the model and full-size drafts

process by which such a problem is solved.

When in the latter part of 1929 I was asked to design bodies for the Marmon Sixteen, I knew next to nothing about the automobile industry or its methods. My first two or three months were spent in intensive study, not of precedent or tradition, but of current engineering and production methods. I was seeking an answer to the fundamental questions one always asks in approaching a new product: "What is it for? How is it made?"

Here was a vehicle to house two hundred horse power, to carry passengers in luxurious comfort over the highways at speeds up to a hundred miles an hour. Since at forty miles an hour, sixty per cent of the engine power is used in overcoming wind resistance, the reduction of wind resistance is obviously the first consideration in automobile design. Some day we shall have tear-drop automobiles but the market is not yet ready for them. In 1929 a great many basic improvements were still to be made.

By the time my studies were completed these possibilities had all revealed themselves, and the design had taken shape. In the first place this job was to be one unit, an organism poured in one mold, not a body set on an unrelated chassis. Hence the sides were brought down to cover the frame, and all viscera, front and rear were covered.

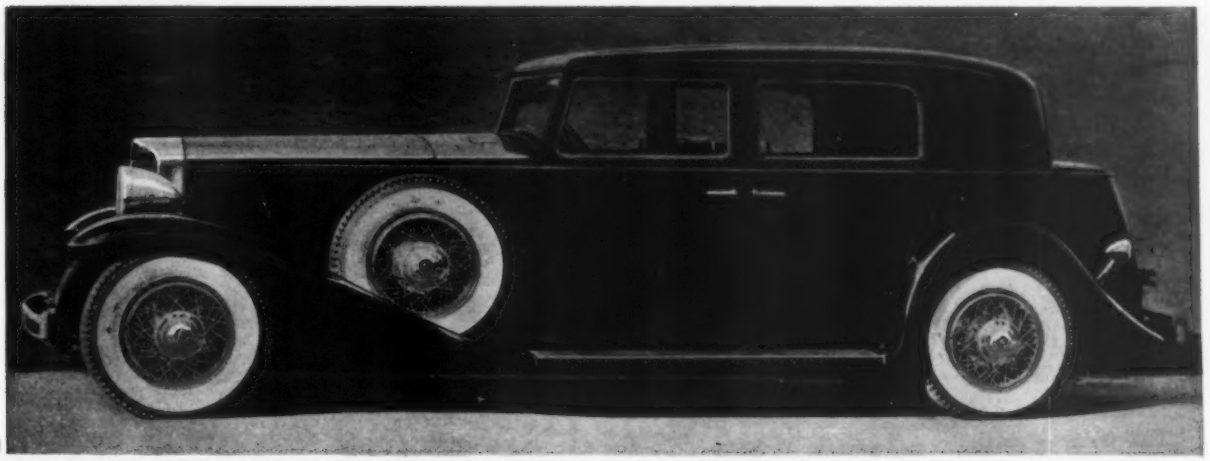
Every detail was subordinated to the whole, and no part was considered except in relation to the rest. Hence mouldings were reduced to a single long thin line from front to rear, all possible details were eliminated.

The silhouette was the thing. This was to be a projectile, moving in a straight line forward. Hence the V-type, slanting radiator, slanting windshield, streamlined fenders, header rolled back to let the wind pass freely over the front (visors were then almost universal), and rear-quarter modelled to reduce suction behind. The latter was further reduced by building on the trunk to carry out the top curve, where length of body permitted. The sides were drawn almost without ogee, like the lines of a yacht, and rear fenders were drawn out as far as the frame allowed. Horizontal lines were emphasized, vertical ones slanted or minimized.

The result not only was fast, but *looked* all its speed and power. To arrive at it, a series of logical steps were followed. First rough sketches, then careful drawings. Then a sixth-scale model was built, perfect in every respect. Then full-size drafts were worked out, and two sample bodies built. From these the various body types were developed.

Throughout this process there was no deviation from the original scheme, no trial and error guesswork. At each stage refinements were introduced, proportions adjusted—at one point I modelled the fenders full size in plasticene to obtain the proper contours. But the first drawings and the model are remarkably close to the finished result. Introduced in November and December of 1930, I have had the satisfaction of seeing almost all the novel features of this design adopted as standard practice for 1932.

It should be easy for an engineer to sympathize with this practical approach to design. His whole background and training prepare him to give it his enthusiastic support. I have never met one yet who, when he realized that the designer had his feet on the ground and was aiming at a perfectly sensible result, would not give him all the help he needed.



In the finished Marmon Sixteen almost every basic feature of the first sketches may be seen: (A) straight lines, simplified moldings, slanting V-type radiator, rolled back header, streamlined fenders, dropped sides; (B) rear end streamlined by building the trunk in a curve continuous with the top; (C) absence of ogee in the sides

THE AUTHOR

Walter Dorwin Teague is design consultant for Eastman Kodak Company, Taylor Instrument Companies, Bausch & Lomb Optical Company, and Corning Glass Works. He is also the designer of the Marmon Sixteen and of products for Thomas A. Edison, Inc., National Radiator Corporation, National Carbon & Carbide Company and others

V-Belt Life, Overload and Pulley Diameter

C. A. NORMAN
Professor of Machine Design, Ohio State University

Recognizing the need for authoritative information on the effect of overload and pulley diameter on the life of V-belts, the editors of *Product Engineering* and of *American Machinist* retained Prof. Norman to carry out exhaustive tests to determine the magnitude of these factors. Here are the results

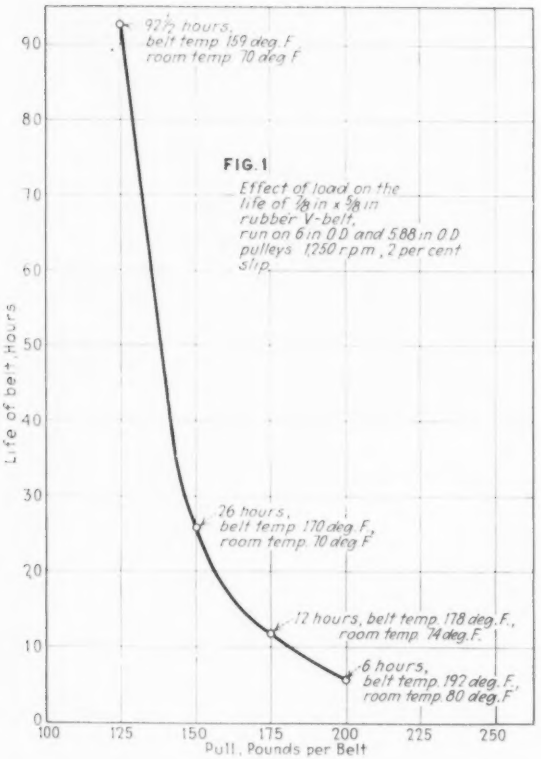

92½ hours, belt temp. 159 deg. F. room temp. 70 deg. F.
26 hours, belt temp. 170 deg. F., room temp. 70 deg. F.
12 hours, belt temp. 178 deg. F., room temp. 74 deg. F.
6 hours, belt temp. 192 deg. F., room temp. 80 deg. F.
Life of belt, Hours
Pull, Pounds per Belt

FIG. 1 *Effect of load on the life of ⅞ in x 5/8 in rubber V-belt, run on 6 in O.D. and 5.88 in O.D. pulleys 1,250 r.p.m., 2 per cent slip.*

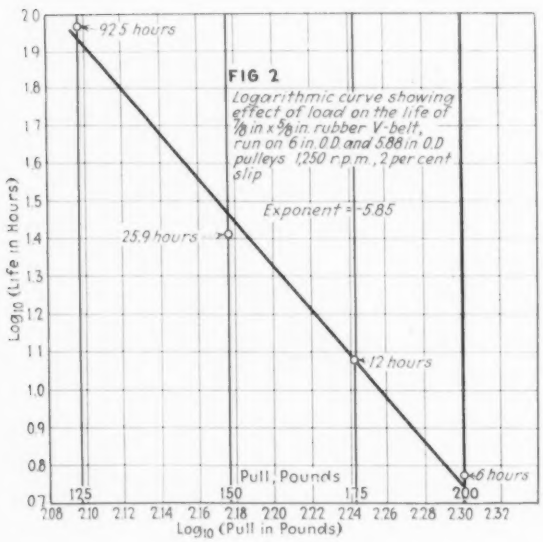


FIG 2 *Logarithmic curve showing effect of load on the life of ⅞ in x 5/8 in. rubber V-belt, run on 6 in. O.D. and 5.88 in. O.D. pulleys 1,250 r.p.m., 2 per cent slip*

With the rapid rise of the V-belt for power transmission, the question of the influence of load and pulley diameter has become acute. The writer therefore welcomed the opportunity offered by the editors of *Product Engineering* and *American Machinist* to carry out life tests on V-belts to determine the effect of these two factors. It so happens that the belts purchased in the open market for these tests are relatively sensitive to both of the factors mentioned, and particularly so to small pulley diameters. There are belts that stand up better, but what the machinery designer must bear in mind is that the majority of the users of his product are likely to replace the belts he specifies for it with any belt of equivalent dimensions that may be locally available. To make his design as nearly fool-proof as possible he should therefore take account of conditions that affect the belt of lower quality most adversely, and make them as favorable as he can. The results of these tests will consequently be useful in defining the effect on belt life of two of the most destructive factors, overload and insufficient pulley diameter.

In order to obtain results within a reasonable time, it was necessary to produce conditions known to affect belt life adversely. The load tests, therefore, were conducted using a pulley diameter about 75 per cent of the minimum diameter necessary for a reasonable belt life. The diameter tests were run under about a 50 per cent overload.

The tests were run on belts purchased in the open market and are thus standard commercial belts, although possibly not belts of the greatest durability available. The machine used is the life testing machine designed by the writer and described briefly in *Product Engineering* for April, page 145, and in some detail in *American Machinist*, Vol. 76, page 360.

The effect of variation of load is shown in Fig. 1. It will be seen that when the load decreases from 200 to 125 pounds per belt, the life increases from 6 to 92.5 hours. By plotting the logarithm of the life against the logarithm of the pull, the exponent of the relation between pull and life may be obtained as the slope of the curve. This has been done in Fig. 2 where the points plotted fall nearly on a straight line, the slope of which indicates an exponent of 5.85. In other words, when the pull on the belt goes up, the life goes down in proportion to the 5.85 power of the ratio of the belt pulls.

It should be understood that this exponent may vary with belt design, and perhaps with such other factors as pulley diameters. Belts with considerably lower exponents have been tested.

In investigating the influence of pulley diameter on belt life, one faces a dilemma. In practice the r.p.m. of the driving and the driven pulley and the horsepower of the drive are usually fixed. If the pulley diameter is increased, the belt speed will increase, and hence, if the horsepower remains constant, the effective pull will be reduced. The life of the belt will then be increased both from reduction in pull and from increase in diameter. To avoid this difficulty, it was decided to run the diameter tests at constant belt speed and constant effective pull. This means a change in r.p.m., however, and it is possible that this change has an influence of its own on belt life. As a matter of fact, tests which cannot here be reproduced show that, for certain types of belts at least, the life may go down almost with the second power of the r.p.m., effective pull and pulley diameter remaining constant. Although this is a serious effect, it is much smaller than the effect of increased pull. Provisionally, allowances for the variation of r.p.m. may be made on the basis just given.

Results of the tests with varying diameter are plotted in Fig. 3. It will be seen from this figure that a dotted and a full-line curve are given between the diameters of 6 and 8 in. The reason for this is shown in Fig. 4. The belts tested were of a wrapped type. After somewhat more than 100 hours of running, the wrapping showed signs of failure and the belts began to stretch at an accelerated rate. They ran, however, for over 200 hours more without breaking. Toward the end of this period the wrapping had been stripped completely off one of them over a considerable distance and the belt was running on the bare cords. Naturally a belt in this condition would never be tolerated in actual power transmission. It is, however, difficult to say with exactness where its useful life should be regarded as ended. From the increased slope of the stretch curve occurring after about 100 hours of running, it is evident that here a progressive destruction of the belt section commences. This might be taken as the end of the useful life. If this is done, the curve of life against pulley diameter proceeds along the dotted line, which is probably the more rational attitude to take. If, on the other hand, the curve is drawn to the last point of the test—at which as a matter of fact the cord section was not yet completely broken—the full-line curve applies.

Fig. 5 shows that the dotted branch in Fig. 3 gives a constant exponent of 5.05 for the relation of belt life to pulley diameter. This means that the life increases with the 5.05 power of the pulley diameter.

While the tests here presented do not show it, it is known from other information as well as from general practice that this relation does not continue indefinitely. A point is reached beyond which a further increase in diameter produces little if any further increase in belt life. It is rather evident that for the belt here tested this point is nowhere near reached at a diameter of 8 in. There are, however, other and more flexible V-belts of the same size with which this is much more nearly true.

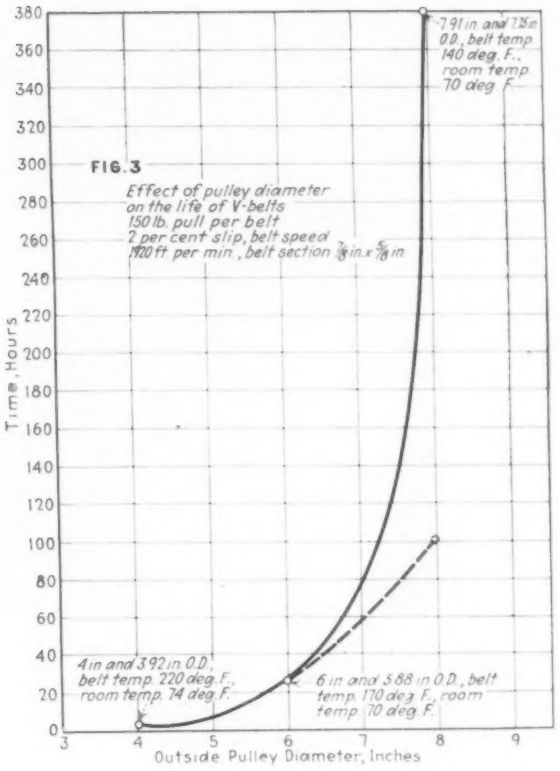


FIG. 3
Effect of pulley diameter on the life of V-belts 150 lb. pull per belt 2 per cent slip, belt speed 1920 ft. per min., belt section ⅜ in. x ⅜ in.
7.91 in. and 7.75 in. O.D., belt temp. 140 deg. F., room temp. 70 deg. F.
4 in. and 3.92 in. O.D., belt temp. 220 deg. F., room temp. 74 deg. F.
6 in. and 5.88 in. O.D., belt temp. 170 deg. F., room temp. 70 deg. F.
Time, Hours
Outside Pulley Diameter, Inches

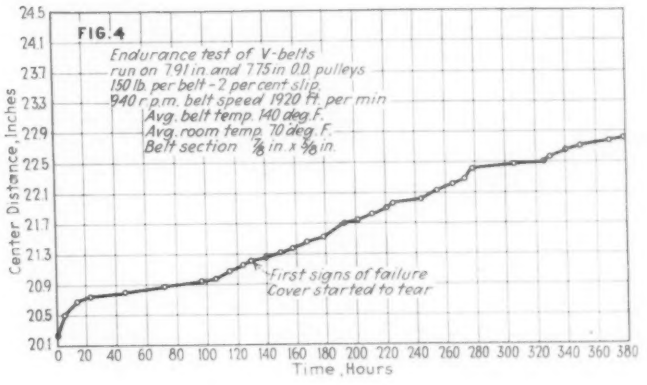


FIG. 4
Endurance test of V-belts run on 7.91 in. and 7.75 in. O.D. pulleys 150 lb. per belt - 2 per cent slip 940 r.p.m. belt speed 1920 ft. per min.
Avg. belt temp. 140 deg. F.
Avg. room temp. 70 deg. F.
Belt section ⅜ in. x ⅜ in.
First signs of failure
Cover started to tear
Center Distance, Inches
Time, Hours

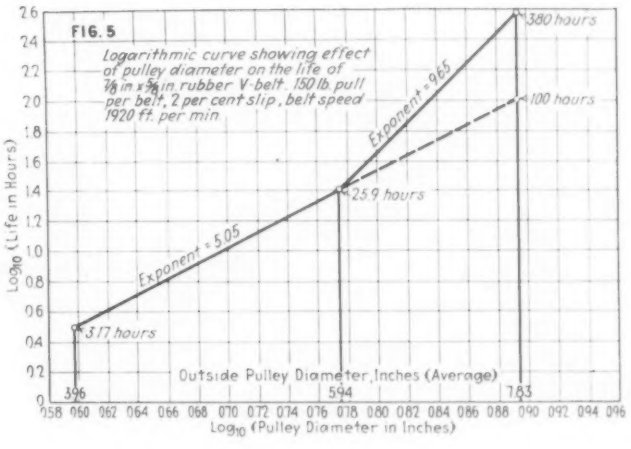
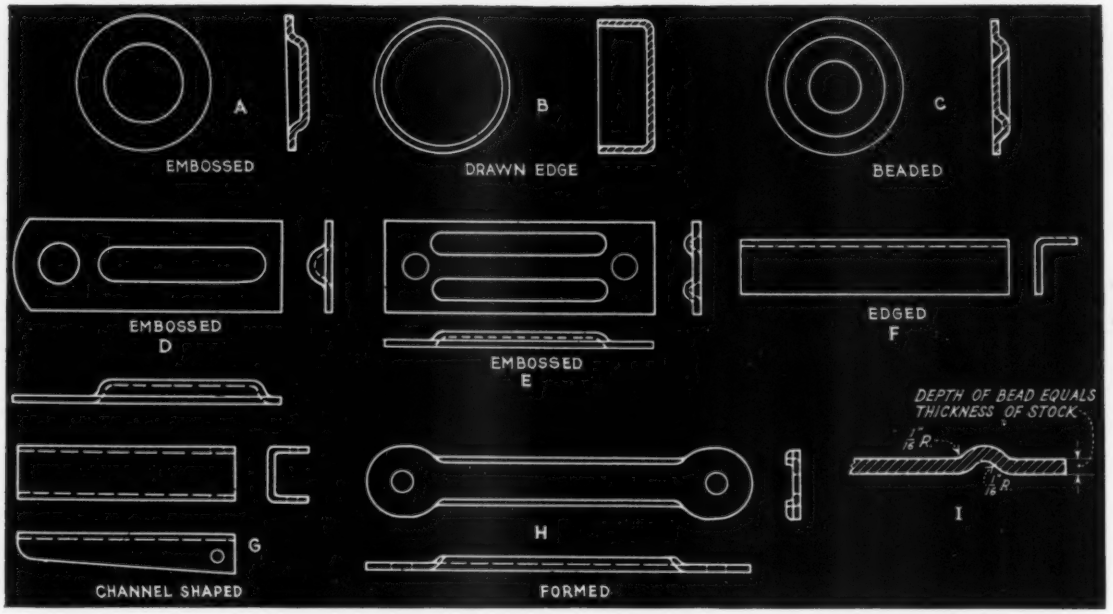


FIG. 5
Logarithmic curve showing effect of pulley diameter on the life of ⅜ in. x ⅜ in. rubber V-belt. 150 lb. pull per belt, 2 per cent slip, belt speed 1920 ft. per min.
380 hours
100 hours
25.9 hours
3.17 hours
Exponent = 9.65
Exponent = 5.05
Log10 (Life in Hours)
Outside Pulley Diameter, Inches (Average)
3.96
5.94
7.83
Log10 (Pulley Diameter in Inches)



I. These bead dimensions are typical an embossing draw is a simple and effective means to save material and to improve appearance. *B.* A cover or cup with drawn-down edges combines strength and stiffness, but it requires more raw material than the embossed disk at *A*. *C.* Another method of disk embossing which serves the same purpose as illustration *A* and improves the appearance of the product. *D.* An embossing may be used in almost any shape of part to add to its rigidity and strength. *E.* Double embossings in most parts increase the strength or stiffness more than the embossing shown at *D* without loss of raw material. *F.* A common bent angle edge is sometimes necessary where the part should be flat. It requires more raw material for the same size than forms *D* or *E*. A bent down edge will help keep the part straight. *G.* Considerably stiffer and stronger than *F* is the channel design, but it requires additional raw material for the same size part. *H.* A formed-up edge provides a simple means for stiffening the part between the ends without increasing the amount of raw material. *I.* These bead dimensions are typical

Economies in Stampings

J. K. OLSEN
Chief Draftsman,
Stewart-Warner Corporation

"There is no doubt that many tons of raw material are wasted by the use of unnecessarily heavy gages. . . ."

▲

In modern metal stamping practice, the cost of raw material is by far the biggest expense item. Hence, any efforts made to obtain the best and most economical usage of material is of prime importance. Where quantities are large and it is possible by the design of the part to conserve material through better layout in the strip, any consequent increase in die cost will be absorbed quickly.

The first consideration in designing a stamping is the thickness of the stock. There is no doubt that many tons of raw material are wasted by the use of unnecessarily heavy gages, where a simple embossing rib, bead or edge would provide the necessary strength and rigidity. Fig. 1 shows a number of examples of the most commonly employed means to keep stock thickness and weight down to a minimum.

What the depth and shape of the bead should be is a common problem. In thin stock, embossings that are too deep may buckle the piece. General practice for light manufacturing of small parts calls for an embossing of $\frac{1}{16}$ in. radius, with the depth equal the thickness of the stock. Of course, these dimensions should not be taken as fixed as there are many factors that may otherwise influence the designer's decision, such as: location of embossing, thickness of stock, and space. In general, a well-rounded-off embossing with a liberal radius should prove satisfactory.

As in many designing problems, there is an economic limit to which thickness reduction can be carried; as the extremely thin gages are approached there is an actual increase in the cost. The minimum economic thickness varies with different materials, but as a practical guide it can be considered as being about 0.020 in. Though there are examples of savings through the use of heavier stock, there are for the great majority of parts advantages in keeping the thickness of stock down to a minimum consistent with the required strength and rigidity.

With embossing, a better product should be obtained. Embossing, ribbing, or similar strengthening results in the saving of raw material without loss of strength and generally with saving in space. In most instances, the complete product may safely be made smaller, lighter and cheaper, besides being improved in appearance.

Another important consideration in designing stampings is the layout of the blanks in the strip. This

varied and complicated problem should be carefully considered by the designer as both the service and cost of the product are considerably affected by the material layout. Wide strips are in general less expensive than narrow ones and should be favored in the placing of blanks.

The first factor to be considered is the quantity of parts to be made. If only a few, ordinary straightline layouts are practical, but if many are to be made, layouts are generally in multiple series in the same strip. Multiple blanks save a considerable quantity of raw material, provided there is opportunity to space and arrange them in whatever manner they fit best into the strip. The necessity for maintaining a fixed grain direction may limit the choice.

Save 10 Per Cent

Double rows of blanks, for example, save at least 10 per cent of the raw material cost of single rows, and can in most instances be stamped as single rows by reversing the strip and feeding it through a single die. With the latter arrangement it may be necessary to increase slightly the stock allowance between blanks and between the blank and the edge, but this increase is negligible. A strip is never exactly parallel between edges; it has also a tendency to buckle and twist when fed through the press, and the increase in spacing provides for this distortion.

Fig. 2 shows the general practice for material allowance between blanks and between blank and edge for both round and square parts. It is best to allow more material where parts are parallel and straight rather than round. These allowances are affected, of course, by the kind of material and shape of the parts, and are only approximate.

Raw material may also be saved by designing to use the waste from one or more parts to make other parts. Very appreciable savings are possible, as the extra parts are obtained at almost no raw material cost. Once it was the practice to call this left-over material "scrap," but in modern practice the possibility of utilizing it should be and is considered. There is, of course, a small scrap value in all waste material, but there is a much larger saving if this can be used for other parts. All that is required is a little foresight on the part of the designer.

Fig. 2—A double layout of blanks saves at least 10 per cent of raw material over the single row and can be punched in a single die by reversing the strip. Approximate figures for the amount of material to be allowed between the blanks and the edge and between the blanks themselves, for both round and square parts are indicated as a fraction of the sheet thickness *T*

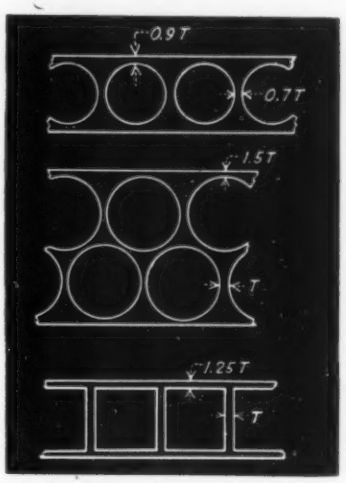
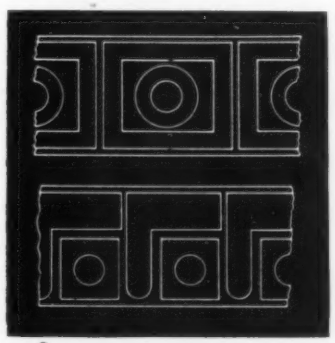


Fig. 4—Large savings can be made by using scrap material for other parts



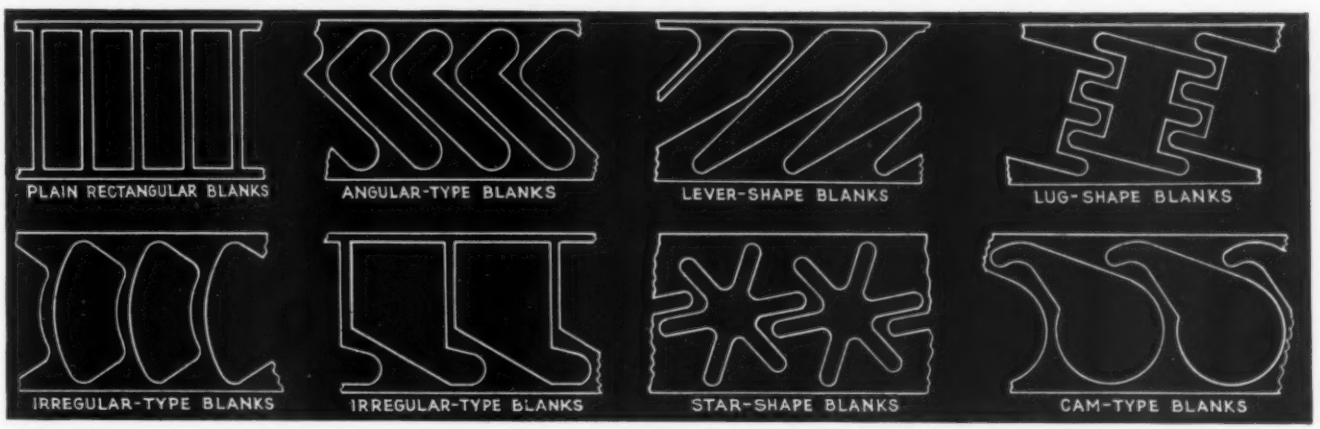
For this reason, it should be one's aim always to concentrate on the least number of gages of material. Besides assisting the production department to keep down inventory, it avoids production hold-ups owing to lack of material of a special thickness or width.

When a stamping has been properly designed and a variety of parts are to be made from the same kind and thickness of stock, it is interesting to see how these various parts can be arranged to produce the best results in the raw material layout. Arranging more than one part in a single strip is many times more complicated than the ordinary single layout, but without exception the savings well repay the designer's added effort.

Often too there is a chance to improve the shape of parts and at the same time obtain a better and cheaper raw material layout. It may seem that many of the savings effected are trivial, but in one million parts, for example, one-tenth of one cent per piece saved will result in a net of $1,000.

Another factor that affects material layouts, particularly for such parts as springs and clinched lugs, is the direction of the grain produced in strip metal by the rolling process. Taking advantage of grain direction serves often to improve the serviceability of the part. Sometimes the advantage can be taken without increasing the raw material, but not

Fig. 3—Typical arrangement of raw material blanks



often. Designs should be carefully studied to see what modifications may justifiably be made. The illustration, Fig. 5, shows two different placements of a lug on two spring pieces, indicating that 20 per cent more material will be used in one design than in the other. The first is to be preferred since the spring can be made from stock the same width as the part, or blanked from a slightly wider strip if required. It is important also to consider the effect of grain direction as regards tension and life of any spring. For fast and continuously moving springs the correct grain direction improves the action and lengthens the life.

Other parts that call for a definite grain direction are clinched parts, seams, and occasionally parts made of a hard material and involving a sharp bend. Since there is less danger of cracking when bent across the grain than with it, its direction should always be considered when a part must be bent. Proper study of this phase of the question will result in fewer rejects and stronger parts. Stiffer and harder materials can be formed more easily across grain than with it. Furthermore, the harder grades of steel are not only stronger but also cheaper. Figs. 6 and 7 show pieces where direction of grain is a consideration.

Fig. 8 illustrates an example in which a compromise is made between raw material cost and the most desirable placement of the piece in the material layout as regards grain direction. The part is a copper clip with four bent lugs made from 0.0508-in. sheet copper. Altering the grain direction slightly in the second layout saves about 15 per cent in material cost. It would be foolish, however, to insist on a rigid adherence to across-the-grain bending in an example of this kind where the grain in the second layout is almost parallel to the lugs.

Of course the whole question of grain direction is intimately tied in with the material that is to be used. Because the consideration of grain direction usually increases the cost, it is usually better to design the part, if possible, so that it will not be necessary to consider the grain direction. As will be shown later, the permissible bend across the grain differs greatly not only with the kind of material but also with the temper.

Recurring Developments

J. A. Covington

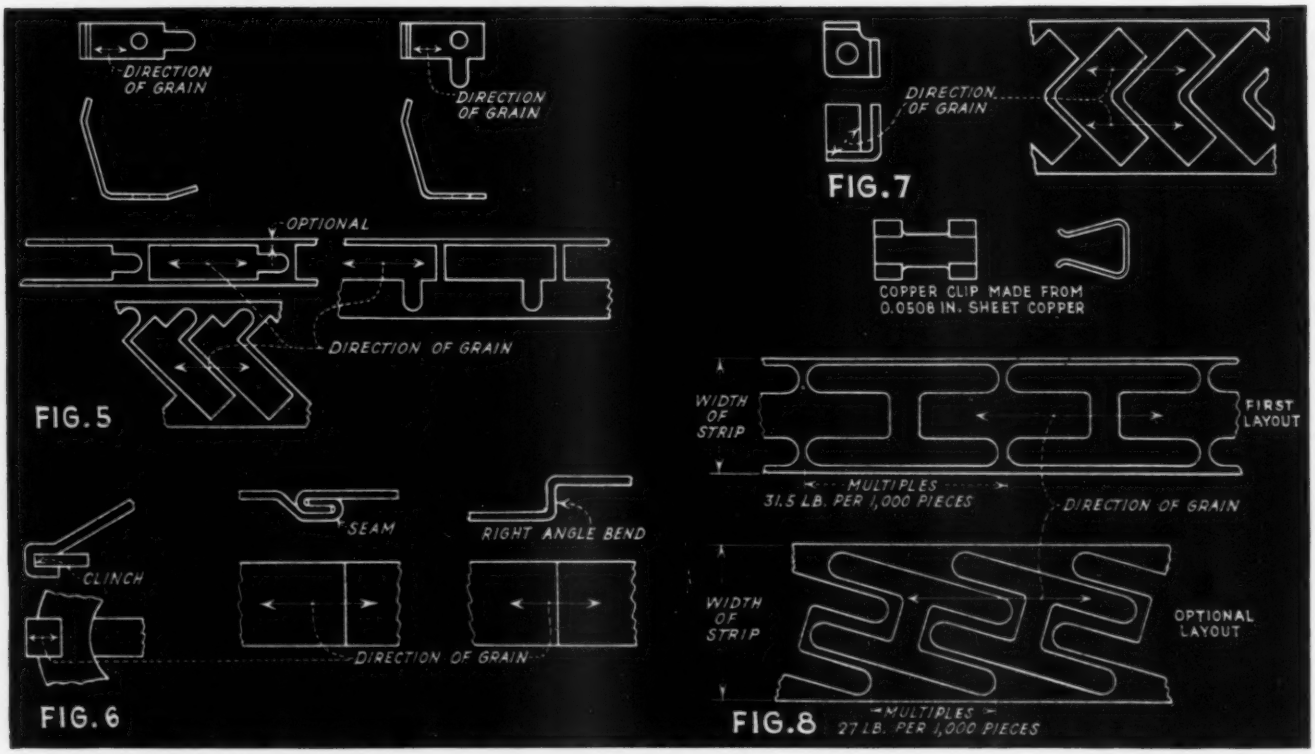
It has rightly been said that history repeats itself, and this also holds true for engineering design.

In engineering design, many belts were replaced by chain drives and gearing, and now we see a return of these belts particularly on short-center drives. Welding made rapid strides, and replaced casting in many fields. Today there is a noticeable trend in some fields back to cast constructions. Worm gear drives were used to a considerable extent two decades ago, to be cast aside because of their low efficiency. Today, the worm drive is finding many new applications.

Why this return to favor? In the instances cited above, belts came back because belt manufacturers improved their product, and by research, improved the design of belt drives. Castings are now replacing welding because of higher quality and a cheaper product. Worm gears are back in favor because of better design and improved manufacture.

One can never tell what the morrow may bring. Eternal vigilance is still the price of success.

Fig. 5—Making use of the direction of grain may increase the amount of raw material necessary. The second layout of the spring with a lug on the side requires about 20 per cent more material than the first, owing to the location of the lug. If there were no grain requirement, the layout could be made most effectively on an angle, as shown. Fig. 6—Three examples showing where the direction of grain can be properly made use of without additional material waste. Fig. 7—For parts designed with lugs formed at right angles to each other, the blanks are placed as close to the 45-deg. line as possible in the strip to compromise the grain direction. Fig. 8—The second, optional layout requires 27 lb. of material per thousand pieces in place of 31.5 lb. in the first instance, a 15 per cent saving that is warranted against any slight loss of strength brought about by this negligible change in grain direction



Criteria for the Design of Bearings

LOUIS J. BRADFORD
Professor of Mechanical Engineering
The Pennsylvania State College

Designers must be in a position to analyze how a given bearing will behave under a given set of conditions. Upon the correctness of the analysis may hinge the success or failure of a new machine. The usual attack upon this problem is to compare the bearing under consideration with a bearing whose performance is known, using as a basis for this comparison one or more factors which exert an influence on the behavior of bearings. The most important of these factors are, bearing pressure, speed of the shaft, and viscosity of the lubricant, the latter being much less important when considerable metallic contact between journal and bearing exists than when the two elements are completely separated by a film of lubricant.

Two Expressions in Use

Two expressions, one based on the relationship of pressure and velocity, and the other relating to pressure, velocity, and viscosity, are in wide use. The first is the equation $PV = C$, and the second is a factor, ZN/P, sometimes called the bearing characteristic. In these expressions P is the bearing pressure in pounds per square inch of projected area, V is the rubbing velocity in feet per minute, C is a constant depending on a number of other conditions relating to the bearing, N the revolutions per minute of the shaft, and Z is the absolute viscosity of the lubricant. Each of these criteria has strong supporters and also opponents, which would indicate that each has a particular field of application in which it gives satisfactory results, but outside of which it is not so reliable. It seems profitable, therefore, to examine these expressions critically in order to determine, if possible, the significance, field of usefulness, and limitations of each.

The relationship $PV = C$ is an empirical one, developed in the days when lubrication was accomplished principally by means of the oil can, sight feed drop oiler, wick, pad of felt, or waste. The more modern methods of forced and flooded lubrication were not in general use, nor were the speeds of the rotating members as high as those frequently encountered today. Although this equation has been extended to apply to these latter classes of bearing by adjusting the constant, it is by no means certain that the relationship between safe rubbing speed and safe bearing pressure is the same for the conditions of imperfect lubrication met with in the first class, and those of complete film lubrication met in the second class. In fact there seems good reason to believe that the relationships are quite different.

The relationship $PV = C$, is in reality, a statement that the work of friction should not exceed a certain value. This becomes apparent if we write:

Work of friction = Force of friction × rubbing velocity = F V

and

Force of friction = Load × coefficient of friction = W × f

Load = Bearing pressure × projected area of bearing = P × A

Making these substitutions we have:

$$PV = \frac{F \times V}{f \times A} = C$$

or

$$FV = C \times f \times A$$
$$= C_1 \text{ (if } f \text{ and } A \text{ are constant)}$$

From the standpoint of form and meaning the equation is almost ideal as a criterion by which to design new bearings of a type for which the value of the constant C has been established. The reason for this is that bearings fail by heating, and heating is the result of the work of friction.

In designing bearings, when should we use the relationship PV = C? Or when should we use ZN/P as the governing factor? Here is a simple, concrete explanation

When a shaft turns in its bearing the work of friction is converted into heat. This raises the temperature of the bearing above that of the surrounding medium and some of the heat generated by friction is radiated or conducted away. If the rate of heat dissipation is less than that of heat formation the temperature of the bearing continues to rise until equilibrium between these actions is established. Then the running conditions become steady. For successful operation the temperature at which this occurs must not be excessive or failure of the bearing by burning out or seizure will occur.

If we assume that the heat radiating capacities of the bearings to be compared are the same, then for equal running temperatures the work of friction should be constant, regardless of changes of pressure and rubbing velocity. This is what the equation $PV = C$ attempts to establish, and what it does establish, if the coefficient of friction remains substantially constant for varying values of pressure and rubbing velocity. Let us see whether this is so.

Coefficient Varies

Figs. 1 and 2, taken from "Elements of Machine Design" by Kimball and Barr, show how the coefficient of friction for imperfectly lubricated surfaces varies with velocity and with pressure. The values of the coefficient will depend also upon the nature and quantity of the lubricant supplied to the bearings and upon the nature of the rubbing surfaces. These curves represent quite accurately, however, the manner in which the coefficient is affected by pressure and velocity.

It will be noted that the coefficient of friction does not remain constant when either pressure or velocity is

varied. And there is little likelihood that the changes in its value caused by a pressure variation will be compensated by varying the velocity in the inverse proportion, that is, by keeping PV constant. It is more likely that when P and V are varied so as to keep their product constant, the coefficient will be different for each pair of values chosen. This means that the work of friction would be different for each set of pressure and velocity conditions, and that the expression $PV = C$ is not an infallible criterion of satisfactory bearing performance. It seems quite probable that the work of friction does not change as rapidly if PV is kept constant as it would if either P or V were held constant and the other quantity varied. In order to hold the work of friction constant the bearing pressure should certainly be reduced when the rubbing velocity is increased, provided the lubrication is imperfect. But the relationship that the pressure should bear to the velocity is not at all clear, and is evidently not expressed by $PV = C$.

When the bearing is completely separated from its journal by an oil film, that is, perfect lubrication, conditions are entirely different from those of the imperfectly lubricated bearing. Here the force of friction is almost independent of the bearing pressure, and varies directly with the velocity, when the viscosity of the lubricant remains constant. A reduction of bearing pressure in order to compensate for an increase in rubbing velocity, would, therefore, have no effect on the work of friction. The bearing would be almost as hot at a light load as it would be at a heavy one. Under these conditions the equation $PV = C$ simply does not apply and is useless as a medium for the comparison of bearings.

The factor ZN/P is almost the complete opposite of $PV = C$. It is rational in origin and rests on certain assumed conditions. In his paper presented at the Spring meeting of the American Society of Mechanical Engineers in 1915, Mr. M. D. Hersey set forth these conditions as follows:

1. The bearing must be in a steady state.
2. The lubricant used in the bearing must be homogeneous.

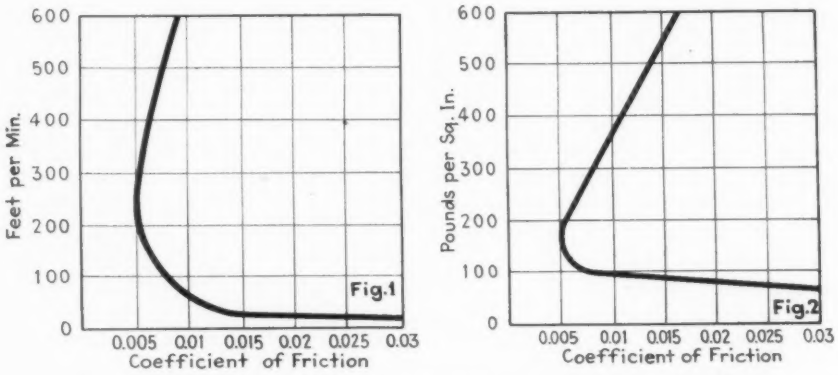


Fig. 1—Variation of the coefficient of friction with velocity of rubbing between journal and bearing for imperfect lubrication. Fig. 2—Variation of coefficient with unit bearing pressure for imperfect lubrication

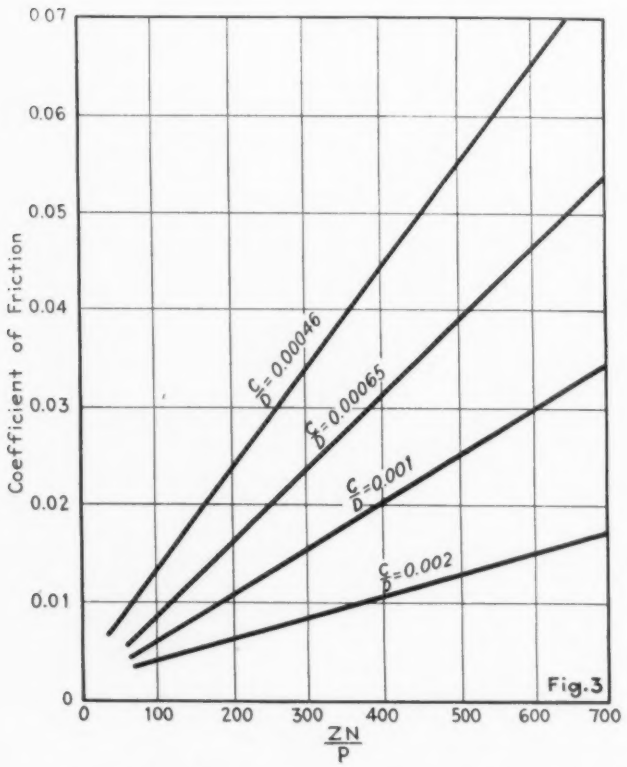


Fig. 3—Coefficient of bearing friction for different ratios of clearance to diameter, plotted against values of ZN/P, for bearings having the same ratio of length to diameter.

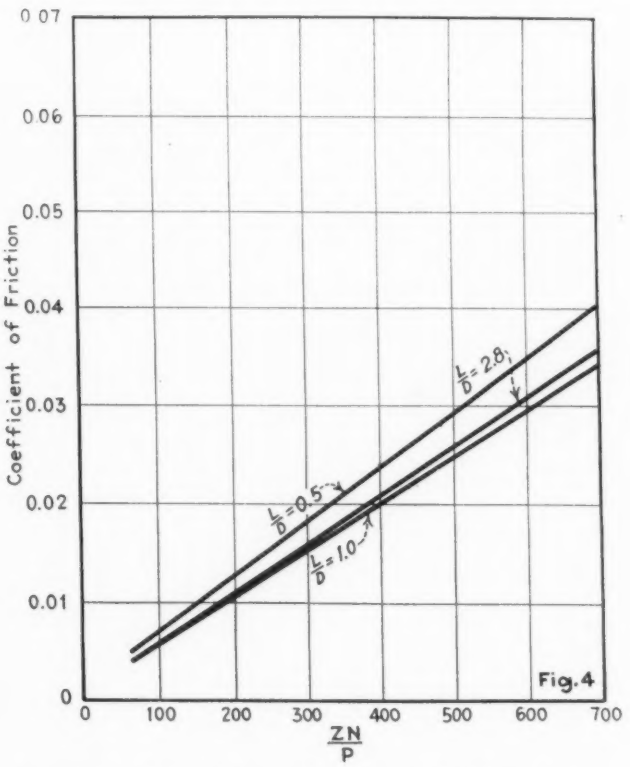


Fig. 4—Coefficient of bearing friction plotted against ZN/P for three bearings all having the same clearance ratio but different ratios of length to bearing diameter.

3. The bearing must be running below the critical speed at which eddy motion would be set up in the lubricant.
4. The effect on the motion of the lubricant of any other forces than hydrostatic pressure and shearing stress must be negligible.
5. The metal surfaces must always be separated by a film of lubricant which is thick enough to have the same mechanical properties as it would have in bulk.
6. There must be no resultant couple acting on the bearing in the plane of its axis.

In addition, the bearings to be compared with each other must be geometrically similar, that is to say, the ratios of clearance to diameter and length to diameter must be the same.

Fig. 3 shows some curves of the coefficient of friction plotted against ZN/P for bearings having the same ratio of length to diameter, but different ratios of clearance to diameter. It will be noted that the various bearings have very different values of the coefficient of friction for a given value of ZN/P. Fig. 4 shows the same curves for three bearings having the same ratio of clearance to diameter, but different ratios of length to diameter. The lack of agreement here, though not so large, is as obvious as it was when the clearances differed. The two sets of curves illustrate clearly the need for geometrical similarity in bearings that are to be compared through the medium of the quantity ZN/P.

Experimental Evidence

Fig. 5 shows a curve of the coefficient of friction plotted against ZN/P for a single bearing operating under a variety of conditions. Loads were varied from 100 to 400 lb. per sq.in. of projected area, and speeds from 500 to 1,600 r.p.m. There were also minor changes in the viscosity of the lubricant caused by changes in running temperature. The values of ZN/P and coefficient of friction were determined and plotted, as shown. It will be noted that all points fall on the curve, thus demonstrating that in geometrically similar bearings, operating under the conditions stated above, equal coefficients of friction will be obtained at equal values of ZN/P, regardless of the values of Z, N, and P entering into this expression. This fact has abundant

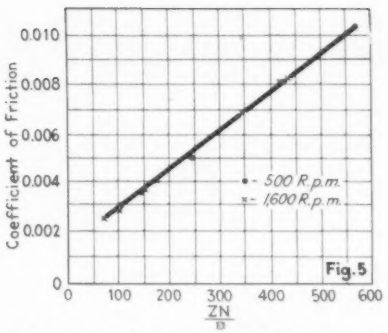


Fig. 5—Coefficient of friction plotted against ZN/P for a given bearing operating under various conditions

experimental evidence in its support.

It should be noted that in the formula ZN/P, Z is the absolute viscosity of the lubricant in the film separating the journal. This is well nigh impossible to determine with a high degree of precision because of the impossibility of introducing any temperature measuring device into a film having a thickness of only a few ten-thousandths of an inch. Using the viscosity of the lubricant corresponding to the temperature of the bearing shell is, however, a fairly good approximation and satisfactory results can be obtained with it.

The quantity ZN/P makes no attempt to take account of the heat radiating capacity of a bearing. Should this be less than the rate of heat generation, the temperature of the bearing will continue to rise and the values of ZN/P will be instantaneous only. Should the deficiency of radiating capacity be great enough the temperature may eventually become so high that the bearing will fail. To insure against this the work of friction should be calculated for the new bearing and compared with the heat radiating capacity at the assumed temperature. Should these agree, the assumed temperature is probably correct. Should there be a wide disagreement a new value of Z more nearly in agreement with the probable running temperature should be taken, and the comparison repeated. In short, while the values derived through the use of ZN/P may be correct for the assumed conditions, there is no guarantee that the assumptions regarding running temperature are correct.

Summing up the discussion given above we may say that $PV = C$ is an empirical equation and suffers from the fault of all empirical equations in that it applies and is safe to use only in the field in which it was developed. The value of C varies tremendously, extremes of 4,500 and 418,000 having been obtained for bearings of different types operating under different conditions. Moreover, due to the way in which the coefficient of friction varies, there is reason to believe that even when bearings are of the same type, C varies considerably if either pressure or velocity is greatly changed. It is wholly inapplicable when the rubbing surfaces are completely separated by a film of lubricant.

The quantity ZN/P, on the other hand, forms an excellent basis for the comparison of bearings that are geometrically similar and perfectly lubricated, provided the value of Z chosen agrees with that actually existing in the film of lubricant in the bearing. However, in order to be assured that the bearing will not fail from overheating, the heat radiating capacity of the bearing must be calculated.

Make It Look the Part

PAUL B. SWINTON

Recently a "sales engineer" gave me some interesting sidelights about welded construction:

"It is pretty hard to figure out what some of these designing engineers expect. They take all kinds of pains to see to it that their castings have nicely rounded corners and generous fillets, and they make the parts have an appearance of strength and durability.

"But when it comes to welded construction, these same designers don't seem to care how the machine looks. The fillet welds are left rough, and what is worse, a tack weld or stitch weld here and there makes it look as if the job were merely stuck together. Then too, the most convenient type of structural member is used, regardless of appearance. The net result is that it looks as if one could kick the whole business apart.

"Designers should call for exposed welds to be ground. Thin edges should have a finishing strip, even if only for appearance. Stitch welding and tack welding should not be used unless it will be out of sight.

"Another factor that has sometimes occured to me is the possibility in the use of extruded sections for welded jobs. These are now available in a great variety of shapes and sizes, yet I have not seen them used much for welded products."

COMMUNICATION AND COMMENT

Long and Short Addendum Gears

EARLE BUCKINGHAM
Associate Professor
Massachusetts Institute of Technology

Much has been written regarding the numerous advantages of long and short addendum gear tooth design as compared with the conventional design. But it must not be assumed that the long and short addendum tooth is a cure-all, and that it can be used to advantage under all conditions. Such is by no means the situation, as can readily be ascertained by a critical analysis of the operation of this type of gear teeth.

In gear design, as with all other design problems, the best solution is often a compromise between conflicting factors. It is seldom that we can take the full advantage of any one factor without losing some part of the value of other factors. The attempt will be made in the following notes to point out some of the advantages of long and short addendum gear tooth design, and at the same time to indicate the nature of the disadvantages thus introduced.

First: When driving pinions of small numbers of teeth are involved, the most obvious advantage of a long addendum tooth here is that it may avoid undercutting. This may or may not result in a reduction of the duration of contact, depending upon the amount of undercut that would otherwise be present. The number of teeth in the mating gear with a correspondingly reduced addendum must be greater enough to avoid undercutting on the gear. Or to put it another way, the amount of increase in the addendum of the pinion is limited by the amount of decrease which is possible in the addendum of the mating gear. Furthermore, the pinion with the long addendum should be made either so that the action starts slightly beyond the base circle or else so that a slight amount of undercut is present. If the driving pinion is so made that it is at the undercut limit and action takes place at, or near, the base circle of the involute, the drive is likely to be noisy because the action at the base circle is seldom, if ever, as nearly perfect as it must be for quiet operation.

Second: A pinion with a long addendum will have a stronger tooth-form than one of conventional design. On the other hand, the gear with the short addendum will have a weaker tooth-form than the conventional.

Third: When the driving pinion has a long addendum, the approach action is reduced and the recess action is increased. The friction of approach is greater than the friction of recess. Hence with such a design, the friction is reduced and the drive is generally quieter. On the other hand, if such a drive is used as a step-up, and the pinion is the driven member, these conditions are reversed so that the friction becomes greater and the drive tends to be noisier than one of conventional design.

Fourth: When a long addendum tooth is used on a driving gear where it is not needed to avoid excessive undercutting, the friction and noise of operation will be reduced but the duration of contact will also be reduced.

All of the foregoing remarks apply equally to all types of gearing.

Fifth: On spiral gear drives, if the driver is made all-addendum and the follower is made all-dedendum, the recess action has a very favorable polishing effect which makes for longer life or greater load-carrying ability and quieter operation. On the other hand, such a design reduces the duration of contact.

Sixth: With all-addendum non-metallic driving pinions, the cutting caused by the corner of the entering tooth of the metal mating gear is avoided and the load may be doubled over that which can be carried by a pinion of conventional design. Here again, number of teeth in the mating gear must be great enough to permit the cutting of an all-dedendum tooth without undercutting, and the theoretical duration of contact is reduced. Because of the relatively large deformation of these non-metallic gears, the actual duration of contact is appreciably greater than the theoretical so that the reduction in the theoretical duration of contact is of but little importance in most cases.

Allowable Loads on Worm Gears

By JAMES I. CLOWER

In response to E. K. Ambler's question in the February number as to the safe load on worm gears, I have with satisfaction used for continuous service condition the formula developed by Bach and Roser. Leutwiler, in his "Machine Design," gives this formula as:

$$W = (m\,t + n)\,b\,s \qquad (1)$$

W = allowable axial load on worm

$$= \frac{\text{torque on worm gear}}{\text{pitch radius of gear}}$$

t = temperature rise of oil bath = 160 deg. F

$$m = \frac{104}{V} + 3.31$$

V = pitch line velocity of worm, ft. per min.

$$n = \frac{305{,}520}{V + 542} - 356$$

b = width of gear tooth measured along arc at root in inches

s = circular pitch in inches.

Another equation that I have used for constant service conditions and which checks fairly closely with the Bach and Roser equation is:

$$W \times V' = 1{,}000{,}000 \qquad (2)$$

W = axial load on the worm

V' = rubbing velocity in ft. per min.

$$= \frac{\pi \times d \times N}{\cos a}$$ and must be less than 1500 ft. per min.

d = pitch dia. of worm

N = r.p.m. of worm

a = helix angle of thread

Still another equation that I have found to be satisfactory, especially for intermittent load conditions is:

$$p\,V = C \qquad (3)$$

p = pressure in lb. per sq.in. on effective area = 70 per cent of the area of two teeth of the gear

V = pitch line velocity of worm, ft. per min.

C = constant = 250.000 for continuous service

In general C for intermittent load conditions may be increased in the inverse proportion which the actual load time bears to the total time. Thus, for one-half time service $C = 2 \times 250{,}000 = 500{,}000$.

Applying equations (1) and (2) to the particular gear mentioned by Mr. Ambler, the following allowable steady loads are obtained:
From equation (1) $W = 2034$ lb. (b was assumed $= 3.125''$) From equation (2) $W = 2000$ lb.

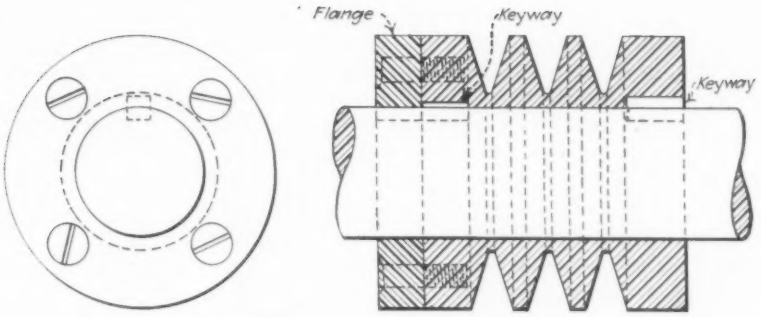
Not sufficient data is available for applying equation (3).

The allowable safe load for worm gears depends upon many factors that are not taken into consideration by the above equations, nor do any of the empirical equations commonly found in handbooks and textbooks consider such factors as workmanship, materials, number of starts, helix angle, included angle between threads, effectiveness of lubrication and character of lubricant used. Undoubtedly, these factors and others have a marked influence.

Mounting Small Sheaves

O. D. Horton

The accompanying sketch shows a method for mounting a small V-type sheave on a shaft of large diameter. Grooves of the proper size and shape for the V-belt are cut in the sheave, after which the sheave is bored to shaft size and short keyways cut in each end. Where the load is heavy it is advisable to use two keys on each end at 90 deg., this giving a stronger construction. Keyways are cut in the shaft to correspond with the keyways in the sheave, the left-hand keyway in the shaft being made twice as long as the corresponding keyways in the sheave, as shown in the sketch. The length of the right-hand key is made a tight fit in the shaft while the left-hand key is only half as long as its keyseat.



Not only is this design particularly suited for small diameter sheaves, but both keys are held positively in position and the sheave cannot creep along the shaft

The right-hand key is put in place and the sheave slid over the key. The left-hand keyway in the shaft now projects beyond the sheave and allows room for inserting the key in the left-hand end. This key is then pushed into place and the flange brought up and fastened to the sheave with the four screws. There being no keyway in the flange this locks the keys and the sheave in place.

I have used this construction on heavy-duty high-speed work and have been able to keep belt speeds to a safe figure, whereas the usual type of construction would have required such a large diameter pulley that the belt speed would have been excessive.

Standard Non-Ferrous Classification

Discussions

Those engineers who have to specify non-ferrous alloys and are frequently confused by a great number of brasses and bronzes differing only very slightly in composition (See "In Behalf of a Standard Non-Ferrous Classification," P.E.—October, 1931, p. 433) will be interested in the following letter from John L. Christie, metallurgist of the Bridgeport Brass Company. It is one of several indications we have received from the field to show that the mills are ready to welcome any attempts to decrease the number of miscellaneous alloys even where it means expense to themselves to do so, if by such elimination they can make specification more simple and cut down the need for large stocks. The letter follows:

"You will be interested to learn that within the past one or two weeks we have taken the necessary steps to eliminate two alloys from our tube mill. Previously we had two brass mixtures which differed from each other slightly in copper content only. These two alloys have been eliminated and another one substituted for the two. It is intermediate in copper content between the other two. It will mean that some customers will receive a slightly lower copper content than they have previously had, and other customers will receive a slightly higher copper content. Having one mixture instead of two permits us to reduce the inventory of metal in process. With only slightly more than half the previous inventory it permits us to make prompter deliveries to customers. It is true that in some cases we will be giving a customer more copper and less zinc than he requires, but it is felt that this slight increased cost to us is more than overcome by the advantages of having just the one alloy.

"In the other case we had two alloys which differed from each other slightly in lead content only. Here again we eliminated both alloys and substituted one with intermediate lead. In this case it was necessary to obtain permission from one customer to ship him tubing with a lead content just outside of his specification limits. The matter was put up to him in such a way that he was agreeable. (There really was no reason why he should not have been agreeable because other concerns making the identical class of article used the composition we proposed). Lowering the lead content will mean that some customers will no longer use the new alloy, but will use another alloy of still higher lead content which has to

be made by the mill for other purposes."

That this mill standardization is in line with what customers want is indicated by the following paragraph from a letter received from D. G. Smellie, director of development and design, The Hoover Company:

"At this time the writer wishes to express his appreciation of the article 'In Behalf of a Standard Non-Ferrous Classification.' We attempted to classify brasses, bronzes, etc., some years ago when we got into very embarrassing situations due to the non-uniformity of trade names and their interpretations. Our efforts along that line were not at all satisfactory. We did, however, write the Non-Ferrous Metals Division of the S.A.E., expressing our hope that something would be done along that line as they had a chance to consider the matter. We have used the S.A.E. iron and steel classifications very advantageously in our business."

Commenting on this same subject, Walter S. Brown describes the experiences of one company. Quoting his letter in full:

"The writer would like to join George S. Brady in his appeal for a standard classification of non-ferrous metals and alloys, as outlined in his article in the October, 1931, number of *Product Engineering*. For a good number of years the writer's firm has purchased practically all its bronze castings from one foundry and machined them in its own plant. The symbols of the Federal Specifications Board were used, such as Bronze M for common brass and Bronze P for phosphor bronze.

"Then came a change in policies and many parts were purchased complete from competitive manufacturers. Some of the bidders inquired what were meant by our Bronze P, and even did not have any knowledge of Federal Specifications Board standardization. Some asked for the equivalent S.A.E. specification and wanted to know if they might substitute the nearest one for those we had specified. The outcome was that it was thought desirable to switch to the S.A.E. specifications for bronzes. But now we are stopped again because these do not include marine bronzes suitable for use with salt water. We require such bronzes but we want to use one series which covers the whole range.

"We recently went over to ordering steels by S.A.E. Specifications and thereby eliminated much confusion together with the removal of the mystery which surrounded the previous use of brand names and meaningless symbols. A national series, on the same lines as the S.A.E. series for steels and some bronzes, could be made to cover all reasonable requirements and would certainly be welcomed by many in the same predicament as the writer's firm."

Reamed Versus Fitted Bearings

FRANK B. FAULKNER
International Nickel Company

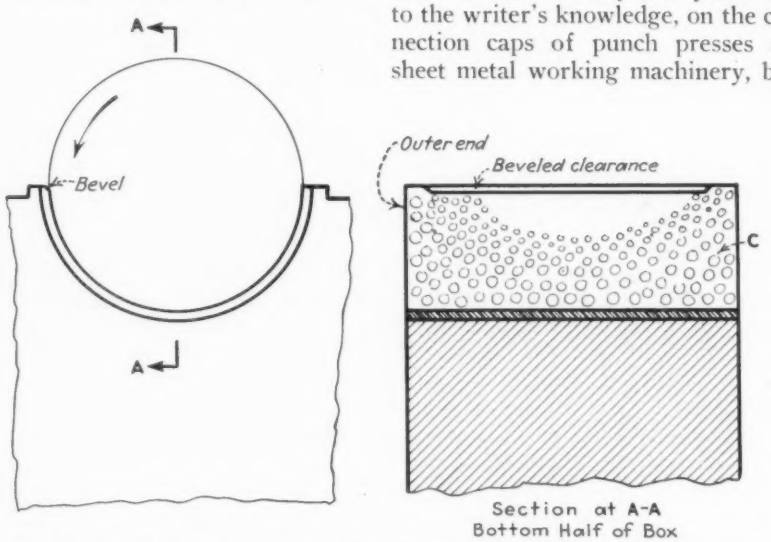
The article by Prof. Bradford in the April number is very interesting and logical. It may be that in most machines, bearings which have been accurately reamed in line are better than those produced by a skilled man with scrapers. Where the bearing is for light loads, reamed bearings are ideal. But for heavy-duty work such as gas engine crankshafts, they would not equal a scraped job.

In one design of gas engine, on which I worked 32 years ago, the main bearings were lined with phosphor bronze shells fastened with copper rivets. The method was to first scrape the boxes to the crankshaft, before the wheels were mounted, being careful to get a good bearing from the center of the boxes to the inner ends on the bottom halves. The caps are of less importance but should have a fair bearing on the outer side to resist the explosion shocks.

The second step was to press on the 2 flywheels and drive the keys. Spinning the shaft and wheels without any red lead (before the days of Prussian blue) gave a very different impression on the bottom halves, for the hard spots showed up near the flywheels, on account of their weight springing the shaft slightly. Now had these bearings been reamed exactly in line, it is certain that 90 per cent of the load would have been on the outer half of the lower box, and the engine would have developed bearing trouble on the testing stand.

The scraping was continued until no area as large as a 50-cent piece was without a spot. Next the edges of the brass liner were beveled as shown on the sketch. The last operation was to scrape a clearance to admit oil on the entering side, but the clearance was for only about 75 per cent of the length along the center, as indicated in the figure.

It will be noted that the bearing hugs the shaft near the ends and thus prevents the oil from escaping, and in this way a scraped bearing is better than a reamed job.



Gas engine crankshaft lower bearing box. The spots indicate bearing contact, the blank space indicating clearance for entrance of oil

Nothing Is New

I. PATRICK
President, Rockford Iron Works

In reading George F. Nordenholt's very interesting article, "Analyzing the Design of the Ford V-8 Engine," I note the description, p. 187, of the design of the bearing cap used on the cylinder block for the crankshaft.

The circular groove principle was used at least twenty-five years ago, to the writer's knowledge, on the connection caps of punch presses and sheet metal working machinery, built

by the Consolidated Press & Tool Company, Hastings, Michigan, now a part of the E. W. Bliss Company.

The writer was appointed chief engineer of the Consolidated Press & Tool Company in 1911, and previous to that date the principle of the circular groove was in use, having been originally designed for use on the Consolidated presses by Mr. Walsh, now president of the Walsh Press & Die Company, 4709 W. Kinzie Street, Chicago. I think Mr. Walsh still uses this or similar construction on the connection caps of the presses built by Walsh Press & Die Company.

If anyone deserves credit for "burning the midnight oil," that credit should be given to Mr. Walsh.

Bills of Material

P. HALL
President, Hall Planetary Company

In making the assembly drawing, should the bill of material be placed on the drawing? We did this some years ago, but for various reasons have now discontinued the system.

We have noticed that some of the prints that are mailed in to us by our customers have the bill of material on the assembly drawing, but these are few compared to the number of prints that come to us without the bill of material on them.

Now, of course, it is recognized that what is convenient and practical in one plant may be entirely out of place in another organization. Different fields of manufacture will naturally require different engineering department methods. However, confining ourselves to drawings for machines and industrial equipment, all such drawings can be treated fundamentally alike.

Undoubtedly, many other engineers have faced the problem of reorganizing an engineering department and establishing new drawing room standard procedure. Many of them have undoubtedly made the change one way or the other with reference to the question of putting the bill of material on the drawing. What are the arguments for and against putting the bill of material on the same tracing with the assembly drawing?

Suggested Standards For Dimensioning Drawings

By E. C. SHANKS

Ever-changing conditions and requirements bring about the need for modifying established practices and methods. Thus, the system of dimensioning that was considered standard practice in days when the caliper was the most accurate measuring instrument of the machine shop, no longer meets the needs of industry.

When machine finishes were first developed to a point where dimensions to one thousandth of an inch became a commercial proposition and resulted in the manufacture of interchangeable parts, it was found necessary to establish the tolerance method of dimensioning. But tolerance dimensions were confined only to those that governed the correctness of the fit between assembled parts.

Following the further development of precision tools, it was found necessary to specify finer tolerances. Then die castings, precision forgings, and similar production processes were perfected, each requiring the use of some degree of limiting dimensions in order to take advantage of manufacturing accuracy.

There exists a need of properly designating the tolerances permitted on all dimensions. It is not considered practical to place limits on every dimension, yet the need for such limits is acknowledged. The old practice was to dimension everything in feet, inches and fractions of an inch, placing the responsibility of securing the correct fits upon the individual foreman. As machines became more complex and the need for accuracy became greater, the use of decimal dimensions was adopted. Later the practice of limiting decimal dimensions came into general use. It now appears that there should be some uniform tolerances to accompany different forms of dimensions. The following simple system is offered as one solution of this problem.

The practice in some factories is to place all dimensions in inches; in others, to start using feet and inches at some particular size. The latter may be anywhere from two to ten feet. The selection of the point of change must be an arbitrary one. In this proposed system the point chosen is six feet, largely because six-foot rules are in common use. Then all dimensions to seventy-two inches will be in inches and portions thereof; all dimensions beyond this amount to be expressed in feet, inches and portions thereof. The reason for making the change is to keep the dimension within the possibility of mental vision, as well as in the simplest form. The dimension 13 in. is readily visualized; as are 53 in., 15 ft. 4 in., 7 ft. 8 in., and 21 ft. 5 in. On the other hand, a dimension such as 137 in. is difficult to comprehend. One must first mentally reduce it to feet and inches in order to obtain an adequate mind-picture of what is indicated. Further, a dimension of 2 ft. 3 in. is unnecessary because it is not an aid to visualization and lacks simplicity.

The foot-mark (′) is not used because it is productive of errors. It is proposed that a script lower-case "*f*" be placed after the portion indicating feet, thus: 9*f*6″. Finally, the inch-mark (″) can be dropped as it is unnecessary. Many factories have already done this because, other dimensions than those in inches being adequately marked, the inch-mark is best omitted. The dimension then becomes, 9*f*6. If preferred, the abbreviation "*ft.*" can be used instead of the "*f*". The number of inches should always be shown to avoid confusion, as: 9*f*0.

The tolerances applying to all feet-and-inch, inch, and fractional-inch dimensions are to be $\pm .\frac{1}{64}$. This gives sufficient latitude for all rough

In response to numerous requests, we will reserve space each month in this department for the exchange of ideas on engineering department management. In it there will be presented communications from engineers proposing standard practices or discussing different phases of department procedure. Readers are invited to submit proposals, discussions and criticisms

machining, is easily measured and is as accurate as necessary on a large portion of the work in most shops. The small variation of $\frac{1}{64}$ will make little difference in the length of shafting, many shoulder lengths and various casting dimensions.

The next finer set of tolerances suggested is ±.005 and is indicated by putting the dimension in thousandths of an inch, as: .625, .750, 3.250, 4.000, 6*f*2.000, etc. Such decimal dimensions as .562, .031, etc., on which these limits apply, shall be carried no further than the three decimal places. Dimensions to four or five decimal places when tolerances of ±.005 apply are meaningless and would defeat the proposed system.

Closer tolerances of ±.002 will then be indicated by using a decimal dimension to four places, or in tenths of thousandths, as: .6250, .7500, 3.2500, 4.0000, 6*f*2.0000, etc.

Thus there are three standard tolerances in the system which provide for the majority of requirements in most shops. They are indicated, respectively, by three easily distinguished types of "flat" dimensions. This is quite different from the practice of using "flat" dimensions that mean almost anything in the range of tolerances .and thrusting the entire matter of working within the requirements upon the foreman overseeing the work. It also givs a better control over the rough dimensions and at the same time sets definite variations from the nominal to which the machinist knows he can work. Briefly, then, a "flat" dimension, in either fractions or decimals, indicates certain definite tolerances by its form.

For those needs which cannot be governed by the standard tolerances, it is proposed to use much the same method as that in general use. It is commonly conceded that the preferred form for expressing tolerances is that of using two dimensions showing the extremes, rather than the form showing permitted deviation from the nominal. That is, $\frac{2.751}{2.752}$ is preferred to $2.750\ ^{+.001}_{+.002}$, the principal reason being that the former is easier for the shop-man to use and no mental calculations are needed.

Obviously it is inconsistent to use four decimal places for dimensions having a tolerance of .002 or more. So, in this scheme, it is suggested that only those having tolerances of less than .002 be allowed four places and that cyphers be added, if necessary, to fill the four places, while those with a tolerance of .002 or more be limited to three places regardless of whether they come out even or not. Examples:

3.7500	.1562	2.652	4.625
3.7510	.1568	2.656	4.628

This will visually classify them as relatively fine or course tolerances.

In using these non-standard tolerances, attention should be given to the arrangement. It is recommended that the "Go" dimension be placed above and the "Not Go" dimension below. On exterior parts the "Go" will be the larger; on the interior measurements the "Go" will be the smaller one. Then the dimension that is reached first in machining will be the first one read. This is another visual aid toward the reduction of shop errors.

The final proposal is for the decimal portion of a dimension to be slightly smaller and above the integral portion; about the size of, and in the position of, the numerator of a fraction, thus: $2.^{001}$. Blue-prints often have small white dots on them similar to decimal points but, if the decimal is arranged this way, there is practically no chance of error.

In conclusion, this plan offers a uniform system of dimensioning drawings which can be adopted in almost any shop without disturbing the present system. It sets up a series of three standard tolerances to be indicated by the form of the dimension. It provides standardized forms for non-standard tolerances which are used when the standard tolerances do not suffice. It gives visual value and classification by presenting them in different standardized forms. Therefore the plan gives greater control, helps to avoid errors and provides a uniform practice.

DEFLECTIONS

SUGAR-COATED TANK CARS

All sorts of things have been sugar coated but the application of a sugar lining to a tank car is a new one to us. Yet the logic of it it is clear enough. Line a tank with sugar crystals and you can ship sugar solutions safely in it. They arrive uncontaminated by contact with metal, and in a form that is much more usable than a solid that has to be put back into solution by the user. Incidentally, this method of shipping is much cheaper.

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HELIUM IN FRANCE

Helium, the non-inflammable gas which until now has been a virtual American monopoly, is reported to have been discovered in abundant quantities near Toulon, France. What this may mean to French dirigible development can only be conjectured.

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FINISHES FOR DURABILITY

Quoting from the *Cowdrey Gazetteer*, a bulletin issued by the C. H. Cowdrey Machine Works:

"For 0.50 carbon steel it has been noted that when left 'rough turned' the fatigue limit has dropped 18 per cent below that for a good polished surface. Even the surface left by a coarse file may produce a fatigue limit 18 to 20 per cent below that for the polished piece; bastard file shows a 14 per cent drop, smooth file 7½ per cent drop; accidental nicks and scratches may drop the fatigue limit as much as 16 per cent."

Moral—keep a smooth face to avoid fatigue.

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SALES RESISTANCE

The tickless clock has been the aim and ambition of horologists and horological engineers ever since they substituted gears and pendulums for falling sand. One such engineer got the shock of his life when an otherwise intelligent woman told him that electric clocks made her nervous because she could not hear them tick.

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GREEN COPPER

Until very recently, Mother Nature has had no rival in the production of a permanent green finish on copper and bronze. Although the color is that of a basic sulphate of copper, laboratory workers have run into difficulties in attempting to duplicate it. Now, however, the Copper & Brass Research Association announces that it is possible to do the weathering of centuries in a few hours. The resultant finish is insoluble and is wear-resistant.

Seven Meetings—Five Papers—Two Shows

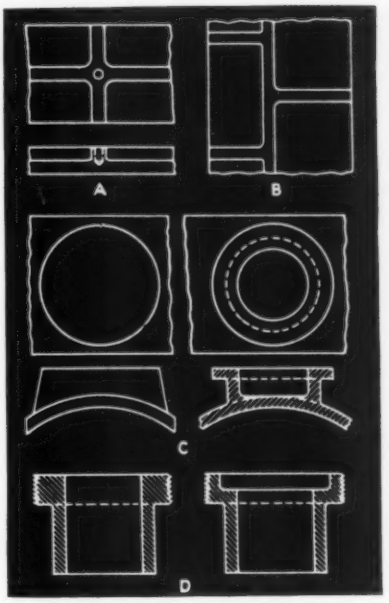
From the combined meetings of American Foundrymen's Association, the Management, Maintenance & Materials Handling Divisions (M-M-M Congress) of American Society of Mechanical Engineers, Foundry Equipment Manufacturers' Association, Gray Iron Institute, Malleable Iron Research Institute, and Materials Handling Institute, Detroit, May 2-5, 1932, there emerge three papers of moment to product engineers and designers: "Influence of Design on Brass and Bronze Castings," by Louis H. Fawcett, metallurgist, U. S. Naval Gun Factory; "Relative Wear of Metals Due to Abrasion," by C. R. Weiss, Link Belt Co., Ewart Works; and "Product Design for the Market," by Robert F. Elder, assistant professor of marketing, Massachusetts Institute of Technology.

Cast Brass and Bronze

According to Mr. Fawcett, there are factors in foundry practice of which one must be aware in the design of brass and bronze castings. A brass with high shrinkage may show low material cost and yet show high manufacturing cost through the excess metal needed for heads and gates and certain necessary additional molding operations. Its chief advantage lies in a fairly uniform grain size both on the outside and inside of the casting. There are even places where a bronze casting may be cheaper than brass, especially where it must withstand water pressure. In general such a casting can be produced with less labor and less excess metal, but the interior structure will be coarser grained than the smooth, close-grained exterior. If considerable machining on the exterior is required, it may make bronze castings unsuitable for hydrostatic vessels.

Where a large volume of metal must surround a small sand core, brass is preferable to bronze. Bronze shows a tendency to "tin sweat" which may penetrate the sand core and cause a hard, irregular surface within the hole. Tin sweat itself is hard and difficult to machine.

The ideal casting is one having the same cross-sectional areas throughout at the time of pouring. Machining allowances may upset such desirable uniformity even though the final finished article may be uniform. When uniform cross-sections are impossible, fillets should be used liberally, yet understandingly, to take care of uneven covering and shrinkage.



In brass and bronze castings: (A) A hole cored in the center of a rib intersection maintains a uniform section at this point; (B) Staggered ribs obviate heavy intersections; (C) The bolting pad at the right has less metal but produces a sounder casting than the solid projection at the left: (D) Uniform section for a threaded collar is attained by additional coring as shown

Metals and Abrasion

Mr. Weiss discloses the results of over 1,400 abrasion tests on a variety of materials, each being compared with key steel, 0.08 to 0.12 carbon, as unity. Experiments determined that a rotating, machined, cast-iron, doughnut-shaped disk, supplied with a fixed amount of a uniform abrasive in the presence of fixed amounts of water, furnished a uniform lap. The disk was rotated at a speed at which centrifugal action would not throw off 80 mesh Crystolon, the standard abrasive selected. It was decided that the net loss in weight under a fixed pressure per unit of area should be the measure of the resistance of the material to the abrasive attack. A circular 1-lb. weight operated directly on top of the metal being tested, each specimen being an accurately dimensioned 1-in. cube which presented 1 sq.in. of projected bearing area. Knife edges so held the specimen that a radius of the lapping disk was parallel with its diagonal. Determinations of loss of weight were ascertained after 3,000 revolutions of the disk.

Cast, work-hardened manganese steel showed a comparative work value of 2.352; wrought iron, 1.021; white chrome iron, 1.495; chilled iron, 1.575; white or hard iron, 1.804; good gray iron, 0.509; phosphor bronze 0.378. Readers are referred to the original A.S.M.E. paper for the complete table which includes many S.A.E. steels in original and heat-treated conditions.

Product Design

"The purpose of the automobile is to furnish transportation; its motivating force comes from the gasoline in the tank. To say that the purpose of the automobile is to encourage the use of gasoline is obviously to put the cart before the horse. Yet the actions of many industrialists seem to testify to the belief that business exists for certain ends of its own and that the consumer's true place in the picture is to provide the fuel which runs the industrial machine." Thus Prof. Elder prefaced his paper.

"Yet as a practical matter of fact it is fairly obvious that the average concern today does not design its product or manufacture it or distribute it on a basis of consumer requirements. This is a sweeping statement, but it is emphasized by the outstanding successes which are every now and then achieved by those concerns which sometimes accidentally and sometimes through intelligent plans, manage to strike responsive chords among consumers. . . . The nature of the manufacturing process too often determines the appearance and specifications of the product, wholly aside from the question of equipment. In most plants the individual in charge of design is more familiar with the technique of production than he is with either abstract principles of de-

sign or the requirements of the product's ultimate users."

Anyone will be repaid for securing the entire paper from A.S.M.E. headquarters and reading Prof. Elder's specific recommendations. After all, as he says, "The acid test of a design, of course, is consumer acceptance."

A.G.M.A. Papers

Quite a group of engineers and designers will wish to read two of the papers presented to the sixteenth annual meeting of the American Gear Manufacturers' Association: "The Design and Use of Worm Gear Hobs," by L. R. Mayo, Brown & Sharpe Manufacturing Co.; "New Light on the Strength of Gear Teeth," by A. B. Cox, gear consultant.

"If we go back even a comparatively few years," said Mr. Mayo, "we find that the tooth shape of a worm thread was usually defined as 29 deg. included angle on the axial section. The use of a straight-sided tooth shape on this section was a natural result of the fact that chasing in a lathe was the almost universal method of production. . . . Due to changing methods of worm thread cutting and to the use of multiple threaded worms, many of which introduce manufacturing difficulties in both the worm and in the corresponding hob if the 29 deg. angle is retained, this tooth shape has been largely superseded. . . .

"In the design of worm gears, the ratio existing between the worm and gear is, at times, fixed. Frequently, however, an exact ratio is not required, and in such a case, the question arises as to which is more desirable—an even or a prime ratio. . . . Using a modern worm having hardened and ground threads, it is possible that the even ratio might not show measurable advantages. From the standpoint of the gear manufacturer, however, the prime ratio is preferable as it automatically produces uniform gear teeth. Regardless of an even ratio, a repeating error is difficult to avoid."

As the author states in his introduction, the various practices and standards advocated by him are those in current use by Brown & Sharpe. He makes it clear that high efficiencies require that helix angles be kept within well defined limits entailing the use of multiple threaded worms. Though the paper is aimed primarily at production, the gear designer will find it informative.

Strength of Gear Teeth

"The standard Lewis method of calculating the strength of a gear tooth," said Mr. Cox, "was presented to the public by Wilfred Lewis in 1892. Because of the relatively great inaccuracy of gear cutting and assembly forty years ago, Lewis assumed the entire load transmitted by a set of gears to be concentrated on a single one of the weakest teeth with the tooth in its weakest position of contact. . . . In general, all gears are much more accurately made now-a-days than they were forty years ago. . . . Most high grade gears now approach mathematical perfection closely enough to make theoretical mathematics usefully applicable to them. It is becoming increasingly recognized that the original Lewis Formula is not satisfactory for modern high grade gears. Part of this recognition has already shown itself publicly in a change from the original velocity factor of $600/(600 + V)$ for the higher grade gears (V = pitch line speed), but the strength formula itself has not been changed. . . .

"Some engineers calculate their gears for safe wear, and do not feel the need of accurate strength formulas, because if a gear is calculated for safe wear it is usually also amply strong."

The author acknowledges the apparent difficulty in devising a formula for the strength of gears on the principle of division of the load between two or more contacting pairs of teeth. How to determine what proportion of the total transmitted load each tooth pair carries seems to be a hard problem, but he digs into it and comes to the conclusion, among others, that "there are peculiar advantages in having substantially 2.0 pairs of teeth in contact—no more and no less—in a set of gears. Such gears will replace any set of gears of any type in a given space and will have greater strength and durability, and will operate consistently quieter and more efficiently than the gears replaced."

R.M.A. Trade Show

As we go to press the Radio Manufacturers Association is meeting in Chicago (May 23-26, 1932) and presenting its "closed" trade show, open only to radio manufacturers, jobbers and dealers. From the editors of *Radio Retailing* we have secured the following preview of technical trends:

Manufacturers, while continuing to produce one or two sets designed for rock-bottom buyers, are endowing quality models with modern improvements calculated to justify higher prices. Trend is to "good little sets —and better big ones." Noise control, a deluxe feature well worth the money, is an ingenious system of automatic volume control which reduces tube hiss and static pickup between stations. Automatic volume control, one kind or another, is almost universally featured. Twin speakers are now in a number of stock consoles, one maker cataloging a "twin" as an optional extra, another raising the ante with a three-dynamic job.

Visual tuning indicators of both the meter and neon types are on the increase. Several receivers boast tone booster circuits which automatically intensify low notes when volume is reduced. This compensates for the relative insensitivity of the human ear to low notes, at low volume levels.

A majority of engineers have designed around the new 2.5 volt a.c. tubes, including R. F. pentodes, altering the industry's tube picture virtually overnight. One leading company takes another tack, standardizing on a 6.3 volt a.c. series. Although most audio systems still make use of power pentodes the new Class "B," or push-push system using twin-grid tubes and mercury vapor rectifiers has cut in appreciably. It now appears likely that power pentodes are eventually to be eclipsed by one new system or another. Use of a special, full-wave grid detector in several lines is noted.

Designers, sensibly, are now playing up to the public's liking for more "cylinders." Relatively few factories have put out smaller models than last year but 11, 12 and 13 tube models are no longer uncommon, and a 19 tube console makes its bow.

Practically all lines now include shortwave equipment. Converters and converter combinations still lead the field in point of number but there has been a marked increase in all-wave models, both "superhet" and t.r.f. Many tune as high as 20,000

kc., in addition to covering the broadcast band, while others are limited to police calls. One manufacturer equips consoles with special sockets into which converters of the same make may be plugged, securing power supply for the external unit in this novel manner.

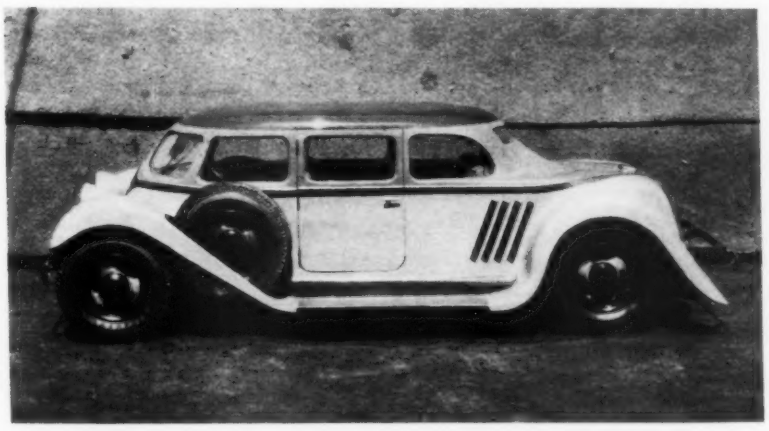
In review: Important advances in tone quality have been made. Class "B" amplification—because of its 10 watts or more of power is instantly available to handle sudden exclamations or musical crescendos—eliminates much "blurring." Larger speakers have been developed to take care of the full volume produced by powerful amplifiers functioning over a wider frequency range. Twin-speakers, better baffles, improved tubes and circuits give added "depth." One popular receiver responds from 35 to 5,200 cycles. The dealer, and his customers, should now be able to hear actually the difference between a new and an old, a low and a higher priced model. The new, quality receivers are really new.

Patent Exposition

Models of more than 5,000 inventions—all of them patented, none of them marketed, few of them marketable—featured the Fourth International Patent Exposition, Grand Central Palace, New York, May 10-28, 1932. Here under one roof were displayed more hopeful, yet hopeless ingenuity, more unconscious humor, more sheer ignorance and ineptitude than in any one place outside the Patent Office itself. Inventions ranged from mechanical fly-swatters to mine hoisting machinery. Inventors came from every walk of life and from nearly every country in the world, each one of them paying twelve dollars per model in the childlike hope of finding a buyer with gold-lined pockets. And not one device in one hundred can or will ever be manufactured and marketed.

Practically without exception the inventions reflected the inventors' fetish of secrecy. It was surprising and sad to see how many of the gadgets had been designed by men entirely out of touch with the fields they desired to invade: a railroad coupler invented by a German tobacco worker; a helicopter by a Pullman porter; a suspender buckle by a spinster. Roughly classified the exhibits ranged from railroad devices to household aids, from toys and games to small tools for shop and home. The number of safety devices of all types and applications indicated a major inventive obsession. Estimated conservatively, ninety-nine of the inventions out of one hundred would never have gotten into the model stage and into the Patent Office if the inventors could have brought themselves to discuss their projects with a competent engineer.

One work-bent old inventor had spent years in perfecting a train of gears "to multiply power"—put in 25 hp. and take out 50 or more—and the glow in his deepset eyes showed forth his faith in his "epoch-making achievement." Another was staking his future on a shovel with a single transverse roller-bearing to "eliminate the friction and work of shoveling."



Though hopeless ingenuities and incongruities featured the Patent Exposition, this finely finished model of a proposed "teardrop" automobile merits attention. Note the three-spoked wheels

No Change In Taper Standards

Technical Committee No. 3 of the A.S.M.E. for the Standardization of Machine Tapers, after considering all phases of the problem, has decided not to recommend any changes at present. It recognizes that the requirements for taper shanks of drills, and those for milling cutters and other tools are not the same and the committee is now considering later developments in sharp tapers with positive holding devices for tools of this kind. The sharp taper of 3½ in. per ft. or 16 deg. 36 min. included angle and with a cam lock worked out by leading milling machine engineers, is to be studied by a sub-group of the committee. For milling cutters and similar tools, this seems to have decided advantages.

Specification for Aluminum Revised

Proposed revision of F.S. No. QQ-A-451 on aluminum has been sent out by the Federal Coordinating Service, Federal Specifications Board, Washington, D. C. The specification covers grades, material and workmanship, packaging, packing and marking, and other information on aluminum ingots, sheet aluminum, aluminum powder for paints, and varnish mixing for aluminum paint. Dr. George K. Burgess, chairman, requests that those who have suggestions to make concerning these specifications communicate with him.

Progress Reported in Lubricant Research

During the past nine months the Lubricants Research Sub-committee, Society of Automotive Engineers, Inc., organized in June, 1931, to develop tests for the evaluation of the properties of "Extreme Pressure Lubricants," and to formulate tests for the determination of requirements of such lubrication for gears and bearings, has made a general survey to determine what information is available on this subject and to outline a program.

It is estimated that the desired information on the most important phase of the work—load-carrying capacity—can be obtained in approximately one year.

Meetings

A.S.M.E.

American Society of Mechanical Engineers — semi-annual meeting, Bigwin Inn, Lake of Bays, Ontario, Canada, June 27—July 1. Calvin W. Rice, secretary, 29 West 39th St., New York, N. Y. Of principal interest is the symposium on the "Use of Models in Engineering," sponsored by the Applied Mechanics Division, and extending through Tuesday and Wednesday, June 28-29. Papers to be given Tuesday are "Use of Models in Aerodynamics and Hydrodynamics," by O. G. Tietjens; "Determination of the Speed and Power of Ships from Model Experiments," H. C. Sadler; "Use of Models Under Centrifugal Force," Philip B. Bucky. Papers on Wednesday under this symposium are: "Model Testing as Applied to Strength of Materials," R. E. Peterson; "Use of Models in Vibration Research," J. P. Den Hartog; "Use of Models in Studies of Stresses in Structures," Geo. Beggs; "Use of Models in Determining the Strength of Thin Wall Structures," Harold E. Saunders and Dwight F. Windenburg. Other papers of interest are: "Design of Anti-Friction Bearings in Railroad Rolling Stock," T. V. Buckwalter; "Developments in the Design and Application of Gearing," Henry E. Merritt.

Aeronautical Division, annual meeting, Hotel Statler, Buffalo, N. Y., June 6-8. Monday, June 6—papers of interest are: "Development in Aluminum Alloys in Relation to Economies in Aircraft Construction," by C. F. Nagel, Jr., technical director, U. S. Aluminum Co., and G. O. Hoglund, aeronautical engineer, Aircraft Development Aluminum Co.; "A Comparative Study of Welded & Riveted Fuselages," presented by Louis C. Rosenstein, president of Philadelphia Aeronautical Society, and C. Bennett, chairman of committee; "Monocoque Aircraft Construction in Theory and Practice," by Richard L. Schleicher, structural engineer, American Airplane & Engine Corp., "Shaft Drives for Airship and Airplane Propellers," by J. L. Goldthwaithe, Allison Engineering Co.; "Torsional Vibration of Aircraft Engine Crankshafts," by W. G. Lundquist, engineer, Wright Aeronautical Corp. Papers on Tuesday are: "The Ideal Transport Airplane," by Ernest W. Dichman, construction engineer, Chance Vought Corp.; "Standardizing Location and Direction of Movement of Stabilizer Control," by Clifford Swartz, General Aviation Mfg. Co.; "Heating and Ventilating of Cabin Airplanes," by M. Gould Beard, American Airplane & Engine Corp., and "The Possibilities of Cabin-Type Autogiros," discussion of which will be lead by A. E. Larsen, chief engineer, Autogiro Co. of America.

Applied Mechanics Division, annual meeting, Yale University, New Haven, Conn., June 23-25. J. M. Lessells, secretary, Westinghouse Electric & Mfg. Co., South Philadelphia, Pa. Twenty-five papers are on the program to be presented in six sessions. Subjects to be covered are hydrodynamics, aerodynamics, stress analysis, thermal stresses, plasticity and elasticity, and vibration. There will also be a joint session with the American Physical Society.

A.S.T.M.

American Society for Testing Materials — 35th annual meeting, Chalfonte-Haddon Hall, Atlantic City, June 20-24. C. L. Warwick, secretary, 1315 Spruce St., Philadelphia, Pa. A round table discussion is scheduled for the afternoon and evening of Monday, June 20, on the subject "Acquisition of Good Data." A prominent group of physicists and research engineers will give eight papers. Tuesday, June 21, a symposium on steel castings will be held in cooperation with the American Foundrymen's Association. Among the papers in this symposium are the following: "General Data on Steel Castings and Their Fields of Application Governed by the Various Physical Properties," C. E. Williams and C. H. Loriz; and "General Principles of Design as Related to Physical Properties, Soundness and East of Production," F. A. Lorenz, Jr. Other papers during the week are "Specifications for the Gage of Sheet Metals in Terms of Quality Control," R. L. Peek, Jr., which outlines a method of specifying the gage of sheet metals to permit estimation of the probability that any portion of a sheet lies within any given limits with respect to the nominal gage. Another paper "Temperature Effects in Elastic and Plastic Deformation," M. F. Sayre, discusses certain temperature effects in connection with the use of elastic-member calibration devices. W. H. Swanger and R. D. France will present "Effect of Zinc Coatings on the Endurance Strength of Steel." In "Fatigue of Shafts Having Keyways," R. E. Peterson will show that the sled-runner key is stronger than the profiled type. H. R. Thomas and J. G. Lowther are scheduled to give "Fatigue Failure Under Repeated Compression." "Effect of Cold Working on the Izod Notched Impact Value of Monel Metal" is written by N. B. Pilling. Lastly, H. K. Herschman and J. L. Basil are the authors of "Mechanical Properties of White-Metal Bearing Alloys at Elevated Temperatures." In all there will be fourteen sessions.

N.E.L.A.

National Electric Light Ass'n—55th convention and exhibition, Atlantic City, N. J., June 5-10. Engineering sessions, June 7. A. Jackson Marshall, 420 Lexington Ave., New York, N. Y., is secretary.

S.A.E.

Society of Automotive Engineers—summer meeting, White Sulphur Springs, West Virginia, June 12-17. J. A. C. Warner, secretary and general manager, 29 West 39th St., New York, N. Y. Among the papers are: "Piston Ring Progress," R. R. Teetor and H. M. Bramberry, Perfect Circle Co.; "Frame Design with Reference to Car Stability," C. R. Paton, Packard Motor Car Co.; "Power Brakes for Trucks and Trailers—Air Type," S. Johnson, Jr., Bendix Westinghouse Auto Air Brake Co.; "Power Brakes for Trucks and Trailers—Vacuum Type," R. P. Breese, Bragg Kliesrath Corp.; "Development of Fixed-Focus Headlighting Equipment," R. N. Falge, General Motors Corp. and W. C. Brown, General Electric Co.; "Free Wheeling Devices and Control," A. M. Wolf, consulting engineer; "Where Do We Go From Here?" F. S. Spring, Hudson Motor Car Co.; "Application of Aerodynamics to the Present Motor Car," H. G. Winter, Briggs Mfg. Co.; "Design Factors and Combustion Control Determine Future of Automotive Diesels," R. P. Ramsey, Buda Co.; "Journal Bearing Characteristics in the Region of Thin Film Lubrication," S. A. McKee and T. R. McKee, Bureau of Standards.

Engineers and their Activities

JUAN DE LA CIERVA, inventor and designer of the Autogiro, will be recipient of the 1932 Guggenheim Gold Medal for outstanding work in the promotion of aeronautics.

FRENCH E. DENNISON has been appointed chief of small machine design, York Ice Machinery Corp., York, Pa.

F. H. DUTCHER has been nominated for chairman, Metropolitan Section, S.A.E., for the 1932-33 term. Other nominations are as follows: A. F. COLEMAN, vice-chairman; F. C. MOCK, vice-chairman for aeronautics; WILLIAM E. JOHN, vice-chairman for marine; C. H. BAXLEY, treasurer; and J. EDWARD SCHIPPER, secretary.

THEODORE M. EDISON, youngest son of the late Thomas A. Edison, has been granted exclusive rights by the United States Patent Office on a device for the elimination of vibration of machinery. This device is said to be applicable to any machine from a phonograph to a large dynamo.

COL. GEORGE H. FENKELL has been chosen to represent the Detroit Engineering Society on the American Engineering Council. He fills the vacancy caused by the death of Major Gardner S. Williams, who represented the Society since 1921.

COL. GEORGE A. GREEN, formerly vice-president in charge of engineering of General Motors Truck Co., is now vice-president in charge of operations.

R. E. W. HARRISON, formerly chief engineer of Cincinnati Grinders, Inc., has resigned as sales engineering director of the Cincinnati Milling Machine and Cincinnati Grinders, Inc.

F. E. Dennison

H. O. Hem

R. E. Peterson

H. S. Woodruff

HALVOR OLSEN HEM, consulting engineer, Toledo Scale Co., is one of the two recipients this year of the John Price Wetherill Medal, awarded by the Franklin Institute. The award was made, "in consideration of the ingenuity shown in perfecting scales of the pendulum type, improving their accuracy, reliability and sensitiveness."

J. C. HUNSAKER has been appointed to serve the Society of Automotive Engineers for a term of four years beginning May 4 as its representative on the Daniel Guggenheim Medal Board of Awards.

F. A. LORENZ, JR., general manager, Industrial Division, American Steel Foundries, Chicago, will present a paper, "General Principles of Design as Related to Physical Properties, Soundness and Cost of Production," at the 35th annual meeting, A.S.T.M., Atlantic City, June 20-24.

CHARLES A. MCCUNE, formerly research director of the American Chain Co., and more recently director of research of the Welding Engineering & Research Corp., has become vice-president and technical director of Industrial Publishing Co. Mr. McCune is past president of the American Welding Society and of the International Acetylene Assn.

RUDOLPH E. PETERSON, manager, Mechanics Division, Westinghouse Research Laboratories, will present a paper, "Fatigue of Shafts Having Keyways," at the 35th annual meeting, A.S.T.M., Atlantic City, June 20-24. At the semi-annual meeting, A.S.M.E., Lake of Bays, Ont., Can., he will present "Model Testing as Applied to Strength of Materials."

R. H. SMITH has been promoted by the Reliance Electric & Manufacturing Company to manage the Development Engineering Department with headquarters in Cleveland.

H. S. WOODRUFF has been appointed engineer in charge of the Design Section of the newly formed Air Conditioning Department of the General Electric Company. Until this appointment he was in charge of design development and production at the General Electric Schenectady Works Laboratory.

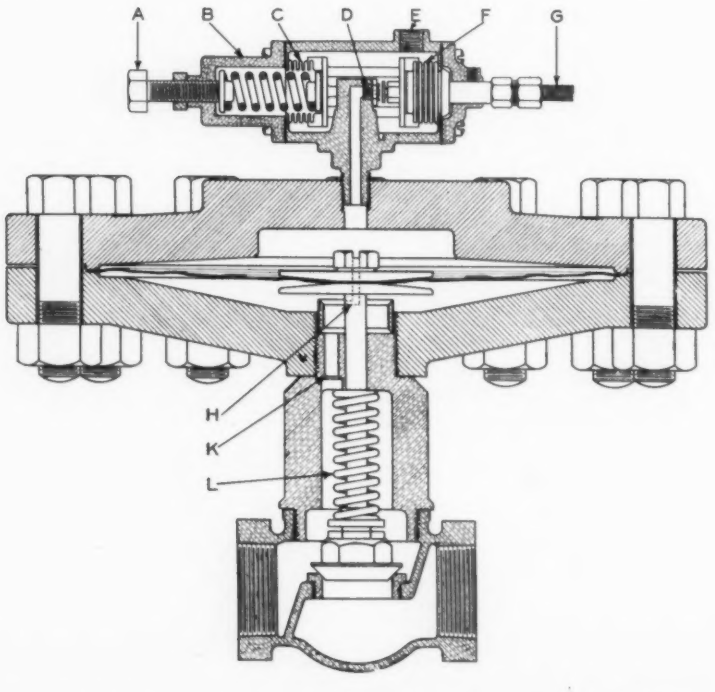
NEW MATERIALS AND PARTS

Tagliabue Steam-Operated Temperature and Pressure Controllers

A new control head provides more dependable and accurate results from the steam-operated pressure and temperature controllers built by C. J. Tagliabue Mfg. Co., Park & Nostrand Aves., Brooklyn, N. Y. This compact head houses the control part as well as the thermostatic bellows, and eliminates leakage and corrosion.

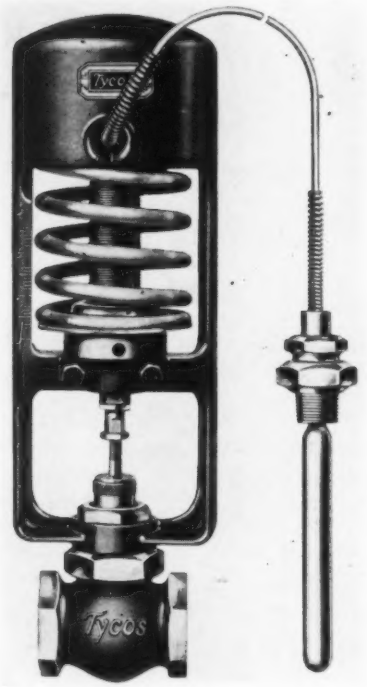
In this controller, pressure of the steam operates a diaphragm valve. Variations in the pressure do not affect accuracy of control as the differential pressure on the diaphragm governs the steam valve opening. Normal position of the steam valve is closed. Steam enters the valve above the seat and passes into the lower diaphragm chamber through the opening *K*, pushing the diaphragm upward and opening the steam valve. The opening *H* in the valve stem allows steam to enter the upper diaphragm chamber. If the steam in the upper diaphragm chamber is not allowed to escape, both chambers have the same pressure and the valve spring *L* will close the valve.

If, however, the steam in the upper diaphragm chamber is permitted to escape through the control port *D*, which has a larger opening than *H*, the pressure above the diaphragm will be less than the pressure below the diaphragm, and the steam valve will be opened. The extent of the valve opening depends on how fast the control port *D* allows steam to escape from the upper chamber. Hence, the steam valve may be wide open, tight shut or in a throttling position. The control port *D* is operated by the thermostatic bellows *F*. *C* is a pressure equalizing bellows.



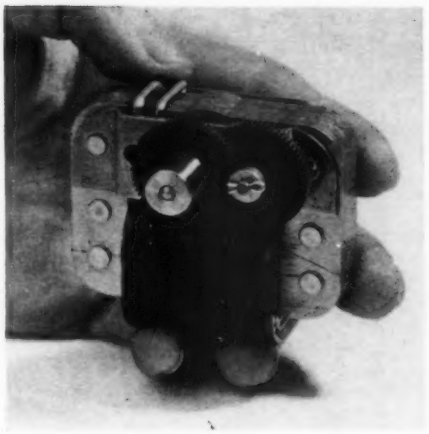
Tycos Self-Acting Temperature Regulator

In the Tycos regulator announced by Taylor Instrument Companies, Rochester, N. Y., friction is claimed to be practically eliminated, resulting in immediate action of the valve stem corresponding to the temperature change. The tube system is also of new construction, being a double seamless bellows with heavy copper connection tubing and small rustless steel bulb with thin wall. The thin wall increases rapidity of response, and the set-up of the system permits minimum volume of actuating medium. The system is also so arranged that the bulb may be heated beyond the upper limit without injury.



"Barcol" Midget Shaded-Pole Motor with Geared-Head

The small shaded-pole induction motor recently put on the market by the Barber-Colman Co., Rockford, Ill., is now available with an integral spur gear reduction. The device will deliver a maximum torque of 2 in.-lb. at approximately 100 r.p.m. Input required is 37 watts or less, depending upon which of the several models of standard motor is specified. This



Maximum torque of 2 in.-lb. at 100 r.p.m. makes this "Barcol" Shaded-Pole Motor with Geared Head suitable for slow-speed domestic machines

geared-head motor finds application in such devices as slow-speed stirring machines, coffee grinders, display turntables, and for winding up heavy spring motors.

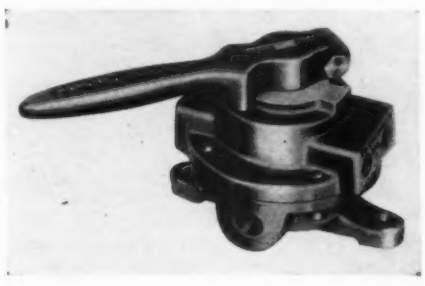
The spur-gear speed reduction provides quiet operation through the use of two steel pinions and two Textolite gears. The second driven gear has a comparatively large hub in which there is an oilless bearing; this gear revolves on the motor shaft. Both intermediate gears are mounted on a stud riveted into the frame-plate. The driven device is to be coupled to the hub of the final driven gear.

Hannifin Duplex Air Control Valve

A duplex air control valve designed for use on machines that employ two air cylinders operating in sequence, is announced by the Hannifin Manufacturing Co., 621 So. Kolmar Ave., Chicago, Ill. The valve is so designed that two double-acting air cylinders can be operated in either direction and in any sequence desired. For this purpose there are four outlet ports.

The valve itself is arranged so that a movement of 45 deg. is sufficient to cause the action desired. For example, starting with the lever at an extreme position, moving it 45 deg. causes the piston of the first cylinder to move forward. Moving it another 45 deg. causes the piston of the second cylinder to move forward. Returning the lever 45 deg. reverses the second cylinder, and moving it back to its original position reverses the first.

The duplex valve embodies all the features of the standard Hannifin "Packless" control valves. It is furnished in two sizes: Model D-37 with $\frac{3}{8}$-in. I. P. connections, and Model D-75 with $\frac{3}{4}$-in. I. P. connections.

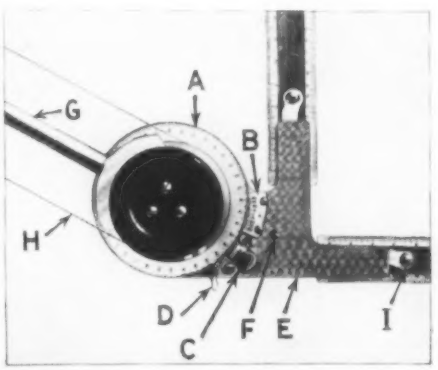


Improved Bruning-Wallace Drafting Machine

An improved drafting machine is now being manufactured by Charles Bruning Co., Inc., 818 Santee St., Los Angeles, Calif. Its maker estimates a saving of 25 to 40 per cent of drafting time, coupled with greater accuracy than when using ordinary methods. The unit may be pushed out of the way to leave the top of the drafting table free. It is ball bearing equipped throughout. The double table clamp is of heat-treated aluminum alloy, while tubes and scale plate are of duralumin. All steel parts are of tool steel, chromium-plated. The $4\frac{1}{2}$-in. protractor is nickel silver, and can be turned in a full circle. It has machine-cut graduations and a vernier reading to 5 min. Automatic stops at 0-30-45-60 and 90 deg. in all four quadrants are provided. Working radius of the standard-size unit is 44 in., or an 88-in. circle. Tube arms are $19\frac{1}{4}$ in. long. The clamp or anchor is designed to be used on any edge of the board.

The small illustration shows a close-up of the protractor head. Two scales seen at right angles to each other revolve with the vernier around the protractor. These stops may be set or released simply by moving knob *F*. *A* is the protractor, *B* vernier, *C* thumb release operating the automatic stop, *D* protractor clamp, *E* scale plate, *F* automatic indexing stop lock, *G* tube with knurled hand grips to adjust protractor and tension of hand, *H* pulley band, and *I* scale adjustment.

Other sizes of drafting machines and left-handed units are supplied as special equipment.



Davey Model S-2 Vibrometer

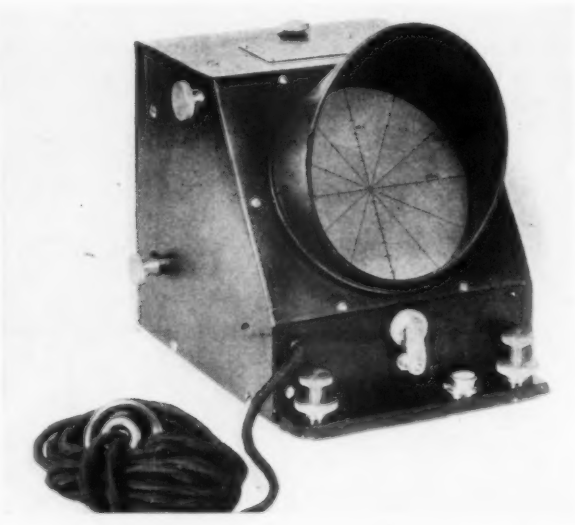
A Model S-2 vibrometer, capable of measuring vertical and horizonal vibration simultaneously, is being manufactured by Electrocon Corp., 6 Varick St., New York, N. Y. The device is set upon the object whose vibrations are to be measured, and a spot of light traces out the exact path of the vibration, magnified many times, on a ground glass scale. The unit will measure frequency and phase position in addition to amplitude if used in conjunction with Davey portable balancing equipment.

The instrument works on the principle of the seismograph, the main element being a weight suspended on springs. When the instrument is subjected to vibration above a certain frequency, this weight will remain inert in space, partaking of no motion from the vibration in either direction. The method of suspension used makes possible the use of a light weight, coupled with a low natural frequency of oscillation of less than 100 per minute. A small concave mirror carried on a spindle is mounted on the weight. Light from a small incandescent bulb passes through a small aperture and is reflected by the mirror in the form of a round spot of light on the ground glass scale. To magnify relative motion between the weight and frame, a silk thread fastened to the frame is wrapped around the mirror spindle. If vibration is in one direction only, oscillation of the spot of light will make it appear as a straight line in that direction. Vibrations equal in both directions will produce a circular track of light, and unequal amplitudes in the two directions will produce an elliptical light track.

The instrument will indicate accurately amplitudes of vibrations of frequencies above 400 per minute up to approximately 6,000 per minute. Good comparative readings may be obtained at frequencies below 400, but to obtain actual amplitude of these vibrations the instrument must be calibrated. The instrument is supplied in two standard sensitivities, Model 2-A, which magnifies vibrations approximately 500 times, and Model S-2B, which magnifies approximately 250 times. Special sensitivities may be supplied

A standard film holder is supplied with each instrument so that photographic records may be obtained of repeating vibration. When supplied with a Neon tube, the instrument may be used in conjunction with the Davey portable balancing equipment.

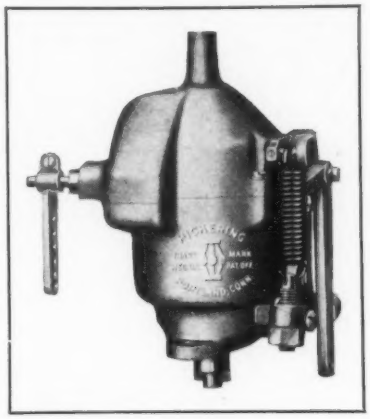
The weight of the unit is 8 lb.; height, 7 in.; length, 7½ in.; and width, 5½ in.



Both vertical and horizontal vibrations are measured simultaneously by the Davey Model S-2 Vibrometer, and a spot of light traces the exact path of the vibration on a ground glass screen

Pickering Type 2,200 Governor for Internal Combustion Engines

Speed regulation within 3 per cent variation is the feature of the Type 2,200 governor developed by Pickering Governor Co., Portland, Conn., for use with internal combustion engines. It is thus adapted to critical service, such as electric generation. The design includes New Departure ball bearings at all stress points to give quick response to engine load changes and durability. The model shown is intended for constant-speed operation, but provision can be made for variable speed or remote control, as required. While primarily intended for gas-engine use, for ratings ranging from 75 to 150 hp., the Type 2,200 has also been used on small Diesel engines. Its power characteristics permit operation of an injection system, and where used in connection with Bosch fuel pumps it has handled up to six in the line.

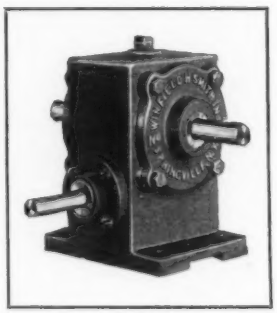


The company also produces a line of smaller governors for 4- and 6-cylinder gas engines of the high-speed type, where the space for installation is limited, and is now developing larger units of the Type 2,200 to take care of Diesel engines larger than 150-hp.

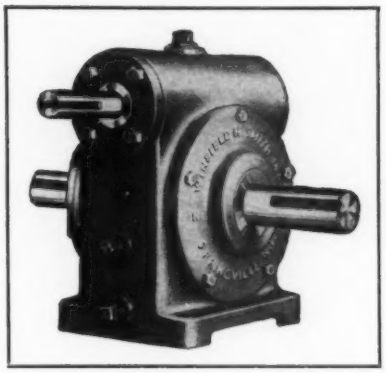
Smith Worm-Gear Speed Reducers

Two series of worm-gear speed reducers have been introduced by Winfield H. Smith, Inc., Springville, Erie County, N. Y., to enable it to offer standard units from 1/20- to 5-hp. output capacity.

No. 3½ B, designed for transmitting loads up to ¼ hp., has a housing with solid top and large side cover plate. This construction permits keying the worm gear to the shaft, and complete assembly work on the worm gear shaft can be made before it is put in the housing. The worm gear is of phosphor bronze mounted on Timken



Model 3½ B

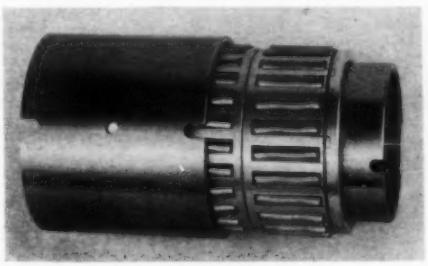


roller bearings. The worm is of hardened steel cut integral with the worm shaft, mounted on Gurney radial thrust ball bearings. All shaft extensions are protected by special oil seals. Standard ratios are 58:1, 36:1, 18:1 and 9:1. A large number of special ratios is also available.

No 4½ T is designed for medium-duty service and has the worm located above the gear. The worm is cut integral with the high-speed shaft, which is mounted on radial thrust ball bearings. Other parts are similar to those of No. 3½ B. Standard ratios and hp. capacities, based on 1,750 r.p.m. of the high-speed shaft, range from 1:6, 1.5 hp. output, to 1:60, 0.25 hp. output.

Bantam Roller Bearing for Printing Press Cylinders

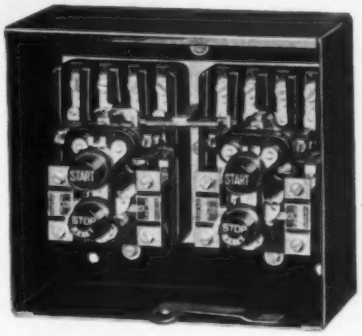
A roller bearing particularly designed for journals on plate and impression cylinders of newspaper printing presses, but suitable for other similar applications also, is being offered by the Bantam Ball Bearing Co., South Bend, Ind. This



type of bearing can be made for very limited space, the usual construction having rollers ½ in. in diameter and races with ¼ in. wall thickness. The rollers and races have a lapped finish, and all rollers and races are made of high-carbon, high-chromium special bearing steel, hardened and heat-treated. One cage has 23 rollers and the other 24, giving a "hunting tooth" effect toward the elimination of synchronized vibration of cylinders at high speed. Each cage operates entirely independently of the other, eliminating bearing "fight" or roller skew.

Square D Class 2510 Across-the-Line Starters

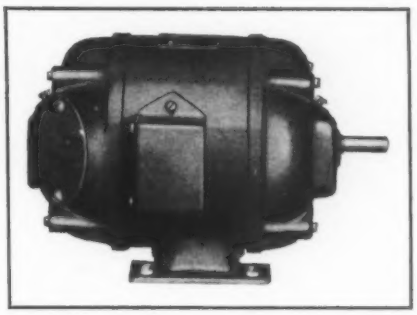
Two-speed separate winding-type and single-speed reversing-type starters for controlling across-the-line motors are recent additions to the line of a.c. manual starters manufactured



by Industrial Controller Div., Square D Co., Milwaukee, Wis. These starters are designated as Class 2510. Both types have general application, but the two-speed type is especially suitable for installations where remote control is not essential. Maximum horsepower ratings of the three-pole starters are 5 hp., 220 volts and 7½ hp., 440–550 volts. Features of the starters are: front operation, thermal overload protection, operation of any relay opens main line to completely disconnect the motor from the line, two quick-break switch mechanisms interlocked to prevent simultaneous operation, and pushbutton reset without replacement of any part after an overload. Maximum dimensions are: 13⅜ in. wide, 10⅞ in. high, and 5½ in. deep.

Century Electric Single-Phase Induction Motors

A new line of brush-riding, single-phase, repulsion-start induction motors has been developed by Century Electric Co., St. Louis, Mo. These are intended primarily for those applications where very quiet operation or higher operating characteristics are

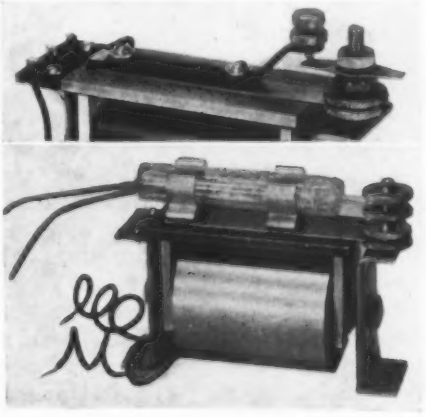


not required. This line of motors is not intended to replace Century brush-lifting motors, which will continue to be made for long-hour duty.

Aminco Supersensitive Relay

A supersensitive relay that will break 1,320 watts with 4 milliamperes, at six volts, that is impervious to dust and dirt, and is safe where an open spark would be hazardous, has been announced by American Instrument Co., 774 Girard St., N.W., Washington, D. C. It will be marketed under the trade name "Aminco" and is designed to control relatively heavy currents through the limited mechanical impulse of thermostat or photocell station.

Relays are made in two styles, in both of which coils, armatures and frames are identical. The relay coil has a resistance of 1,300 ohms and operates on 4 m.amp. at six volts d.c. Relays can be provided for special purposes with coil resistances up to 50,000 ohms. The relay shown at the top, for example, with open platinum-iridium or silver contacts, will break 50 watts. The relay shown below, with a Burgess vacuum contact, will break a maximum of 1,320 watts. Contact is made and broken in an exhausted and sealed transparent glass tube.



New Books and Publications

AMERICAN MACHINISTS' HANDBOOK

[*Fred H. Colvin and Frank A. Stanley*]. Fifth Edition. Thoroughly revised and enlarged. 1135 pages, 4x7 in. Leatheroid covers. Published by McGraw-Hill Book Co., 330 West 42nd St., New York, N. Y. Price, $4.00.

This standard work has been revised and brought up to date. It reflects combined efforts by the authors and a number of well-known engineers toward preparation of a volume that adequately takes into account the changes in shop practice and shop standards that have come into being in greater frequency during late years.

Many examples can be cited tc show the improvements in this edition For example, screw threads have been revised and new material included. Welding has been enlarged and brought up to date. New gear tooth forms adopted by American Gear Manufacturers' Association are included in the section on gear data, and considerable information on nonmetallic gearing is also supplied. Machining methods, which the designer should keep abreast of, are much expanded, one section having an additional 100 pages. Still compact, this fifth edition provides its possessor with wide knowledge of the metalworking field in convenient form.

HOYER-KREUTER TECHNICAL DICTIONARY

[*Dr. Alfred Scholmann.*] Three-language dictionary. Sixth Edition. Three volumes; 2,295 pages, 7¾x11 in. Published in Germany (Hoyer-Kreuter); sole agent in the United States, The Industrial Press, 148 Lafayette St., New York, N. Y. Price, for the three volumes, $55; for two volumes, $37.50; for one volume, $19.50.

One-hundred thousand carefully selected technical terms embracing various arts, industries and sciences are included in this three-language dictionary covering: German-English-French; English - German - French; and French-German-English. Technical elements of the three languages thus become interchangeable. Earlier editions of the dictionary were specialized and limited to distinct fields, whereas the new dictionary is universal in scope and includes 44,000 additional terms. The dictionary serves a two-fold purpose—technical reading and research, and assistance in international trade, both in translating foreign communications accurately and in writing letters. Cooperation was extended to the author by the Assembly of German Scientific and Engineering Societies.

Belting

[*The Garlock Packing Co., Palmyra, N. Y.*]. The 24-page booklet "Garlock Transmission Belting" gives data for designing flat rubber belt drives, information on selection of belts, care of belts, weights and list prices.

Chain and Sprockets

[*Whitney Mfg. Co., Hartford, Conn.*]. Catalog V-100 describes the line of silent chains and sprockets, and also includes engineering data.

Circuit Breakers

[*Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.*]. Two eight-page, illustrated circulars show the company's new line of "De-ion" circuit breakers. Circular 1939 contains a general description of the new breakers, breaker ratings and their applications. Circular 1937 gives construction and application of the 225-amp. breaker.

Clutches

[*Conway Clutch Co., 1543 Queen City Ave., Cincinnati, Ohio*]. Bulletin K-32, superseding Bulletin No. 35, describes the company's cam one-revolution clutch, giving dimensions and descriptions of models with: (1) extended lever, (2) mechanism only, (3) for shaft always running, and (4) cam knockout type.

D. C. Motors

[*General Electric Co., Schenectady, N. Y.*]. Direct-current motors, Type B, constant-speed, ball- or sleeve-bearing, are described in publication GEA-1542. These motors are shunt or compound wound.

Lacquers

[*Roxalin Flexible Lacquer Co., 105-109 Seventh Ave., Long Island City, N. Y.*]. A recent bulletin gives information on three lacquer products: first, a black flexible lacquer for one-coat applications on clean metal; second, aluminum powder finish, and third, a clear lacquer for outdoor exposure.

Lock Washers

[*Shakeproof Lock Washer Co., 2501 North Keeler Ave., Chicago, Ill.*]. Shakeproof lock washers and their uses are described in a new folder.

Oil Seals

[*Aetna Ball Bearing Mfg. Co., 4600 Schubert Ave., Chicago, Ill.*]. "They Shall Not Pass" is title of a 16-page booklet on oil seals, giving diagrams of applications to numerous classes of machinery. Tabulations of sizes with complete dimensions are included.

Switches

[*General Electric Co., Schenectady, N. Y.*]. Bulletin GEA-19G, superseding 19F, describes Types CR-7006-D7B and D7F magnetic switches for a.c. motors. Bulletin GEA-1565, superseding 83A, 813 and 844A, describes Type CR-7009 magnetic reversing switches for a.c. motors. GEA-1587, superseding 1419, describes Type CR-7008 combination magnetic switch for across-the-line starter for induction motors. Bulletin GEA-1588 describes Type CR-7006 explosion-proof magnetic switches for Class I, Group D, hazardous locations.

V-Belts

[*Gates Rubber Co., Industrial Div., Denver, Colo.*]. Bulletin 38-50, superseding Bulletins V-50-100 and V-32-100, gives sizes, prices and weights of "Vulco Rope" drives and lists the interchangeable belt sizes.

Wear and Lubricant Tester

[*Timken Roller Bearing Co., Canton, Ohio*]. The bulletin, "Announcing the Timken Wear and Lubricant Tester," describes a device for obtaining accurate information on load carrying capacity of lubricants, measurement of friction, and wear characteristics of materials.

The Positive Roots of Any Quadratic Equation

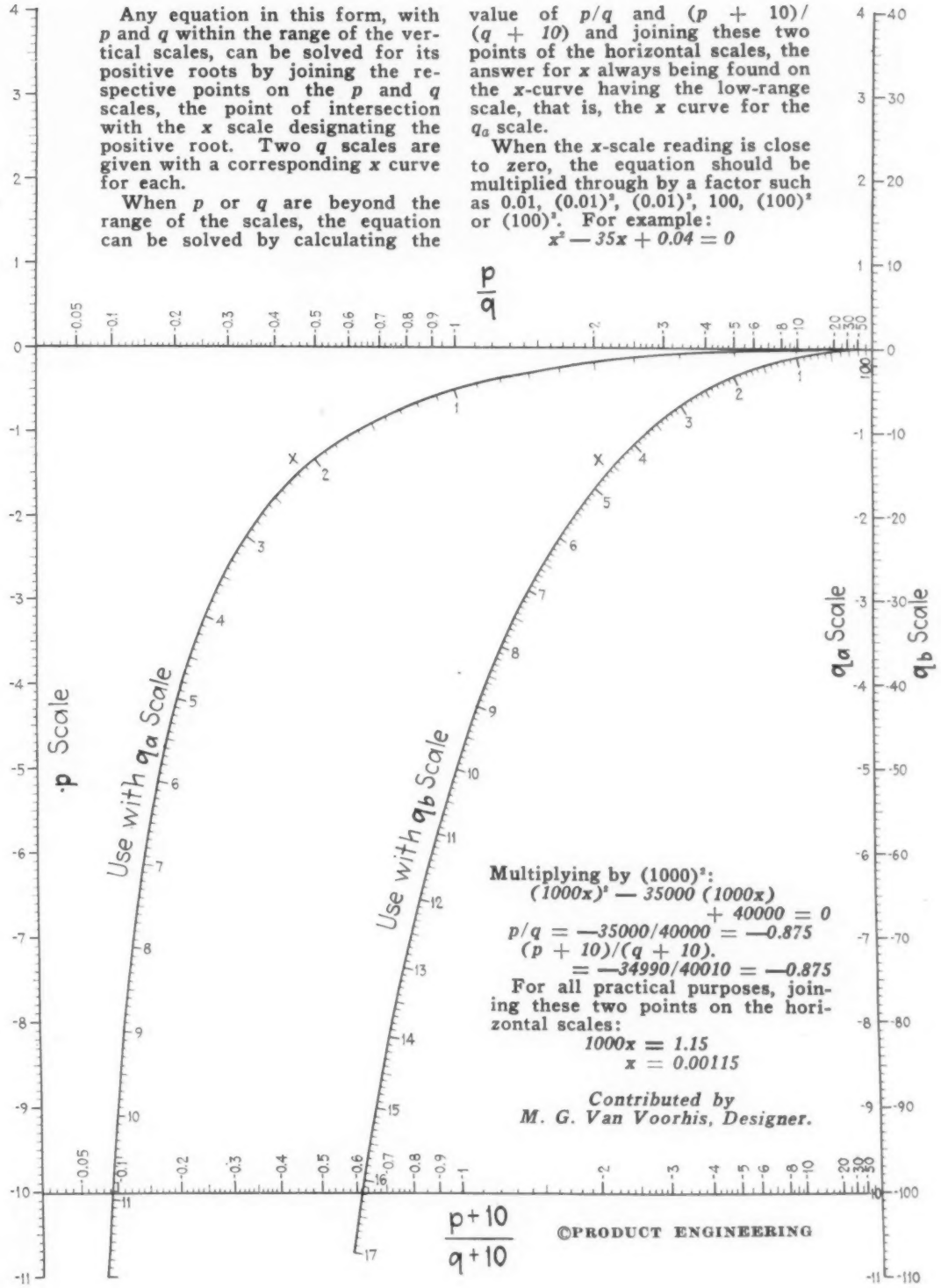
$$x^2 + px + q = o$$

Any equation in this form, with p and q within the range of the vertical scales, can be solved for its positive roots by joining the respective points on the p and q scales, the point of intersection with the x scale designating the positive root. Two q scales are given with a corresponding x curve for each.

When p or q are beyond the range of the scales, the equation can be solved by calculating the value of p/q and $(p + 10)/(q + 10)$ and joining these two points of the horizontal scales, the answer for x always being found on the x-curve having the low-range scale, that is, the x curve for the q_a scale.

When the x-scale reading is close to zero, the equation should be multiplied through by a factor such as 0.01, $(0.01)^2$, $(0.01)^3$, 100, $(100)^2$ or $(100)^3$. For example:

$$x^2 - 35x + 0.04 = 0$$



Multiplying by $(1000)^2$:

$$(1000x)^2 - 35000\,(1000x) + 40000 = 0$$

$$p/q = -35000/40000 = -0.875$$

$$(p + 10)/(q + 10).$$

$$= -34990/40010 = -0.875$$

For all practical purposes, joining these two points on the horizontal scales:

$$1000x = 1.15$$

$$x = 0.00115$$

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M. G. Van Voorhis, Designer.

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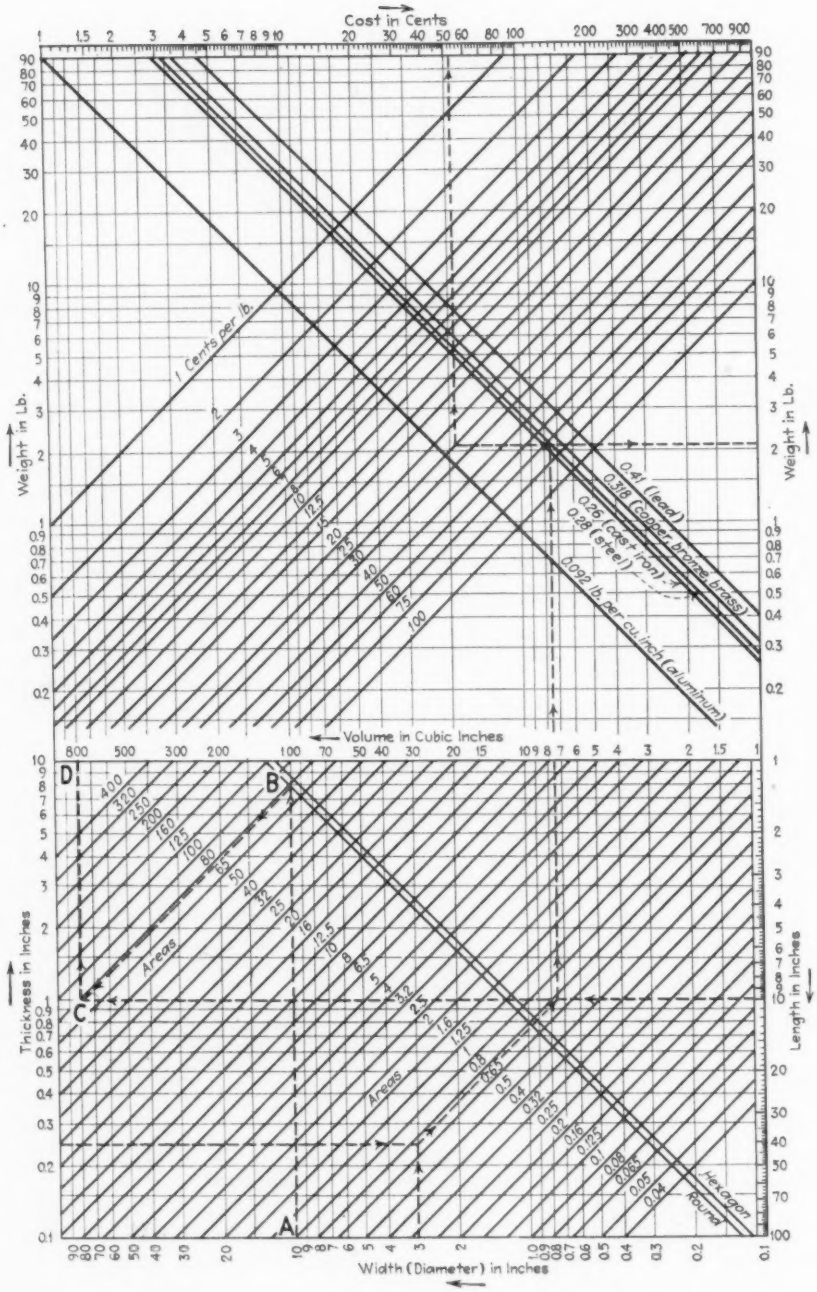
REFERENCE BOOK SHEET

Area—Volume—Weight—Cost



In the examples shown, a thickness of 0.25 in. times a width of 3 in. shows an area of 0.75 sq.in. as designated by the diagonals. Following the diagonal to the horizontal line corresponding to a length of 10 in., the volume is shown as 7.5 cu.in. on the volume scale. Continuing vertically upwards to the diagonal representing 0.282 lb. per cu.in., the total weight is given as 2.1 lb. Moving horizontally to the left to the diagonal representing 25 cents per lb., and then going up vertically, the cost is given as 52 cents.

The diagonals marked "Hexagon" and "Round" are for bars, the bottom horizontal scale designating the diametral dimension and the volume being obtained as indicated by the line *A-B-C-D*, which is for a 10-in. round bar, 10 in. long. Hex bars are measured across the flats.

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